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THERMAL DISTORTION OF VENTILATED BRAKE DISCS

by

M. B. Gerrard

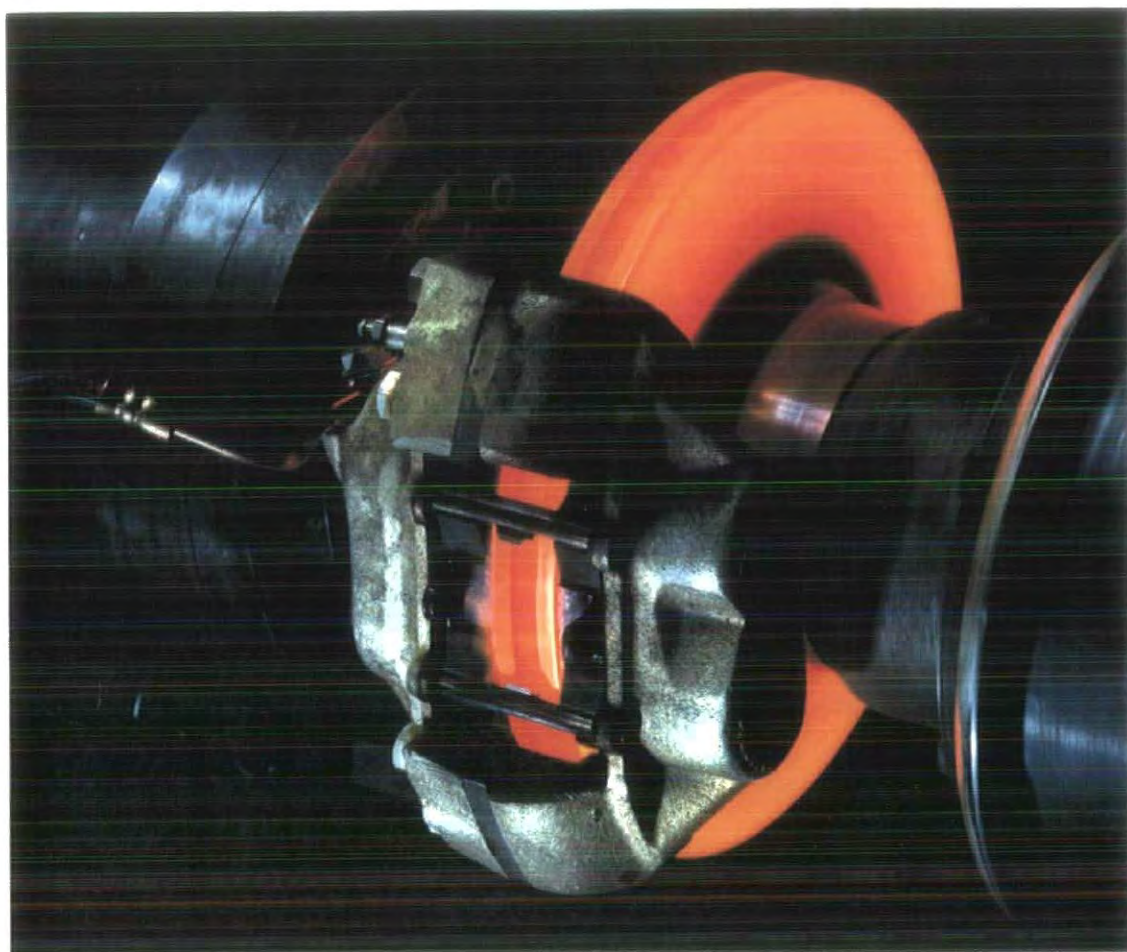
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A thesis submitted to the Faculty of Science
at
University of Durham
for the degree of Master of Science

October, 1993



15 JUN 1994



A ventilated brake disc glowing whilst undergoing testing
on a dynamometer at the Ford Research and
Engineering Centre.

(Ford Photographic Services)

Summary

The current trend in the motor industry is towards faster and more powerful vehicles and this, combined with the necessity for improved suspension performance and vehicle refinement, has meant that the brake disc is required to cope with increasingly higher braking loads whilst being as light as possible. At present, few established guidelines are available to the component engineer to aid the design of an optimised ventilated brake disc. The effects of disc geometry and proportion on the temperatures, stresses and deformation experienced by the disc during braking are not widely known.

It was felt that research into the precise causes and mechanisms of brake disc thermal deformation would allow a detailed understanding of the effect of disc proportion on its performance. This would allow the designer to create discs for particular performance requirements with minimum disc mass and greatly increased confidence in the response of the disc in service. Component and vehicle performance would be improved coupled with a reduction in the time and cost of prototype testing. Shorter lead times and lower piece costs would also follow.

Finite element modelling was employed to provide comparisons of different types of disc design, each one being evaluated principally in terms of the deformation it underwent during a simulated single stop test. Both thermal and mechanical distortions were considered during the simulations in addition to pad contact pressure changes.

This report proposes recommendations on the design of the disc cross-section for the reduction of distortion, in addition to presenting detailed hypotheses concerning the reasons for and mechanisms of disc coning and thermal waving. Conventional and 'inside-out' disc designs were compared and their respective merits evaluated. The importance of a uniform pad pressure is also highlighted and pad design is considered briefly. The report also contains an example of an improved disc design, created using the guidelines developed and its performance is evaluated against current designs of ventilated disc. This disc design combines the reduced coning of the 'inside-out' design with the preferable cooling ability of the conventional design, a significant improvement being achieved over current designs.

Acknowledgements

I am indebted to a number of people for their help in producing this work. In particular, I wish to thank Dr P.A.T. Gill for his supervision of this project. I am very grateful for his guidance which was always provided as soon as requested but never forced upon me. I am especially grateful for his confidence in allowing me the freedom to plan and carry out this research in the way that I chose.

I wish to thank Tom Lawson, Gareth Talbot and Alan Surrey of Ford Motor Company Ltd. I am very grateful to Tom in the first place for having confidence in my research proposal and for subsequently arranging the funding of this work. I owe many thanks to Alan for his support in providing the component data essential for the modelling and to Gareth for conducting experimental work in support of my simulations.

I am also indebted to the staff of the University Computer Centre (especially the operators) for their help with both the main body of research and the production of this thesis. Their response to my frequent demands for help, advice and print-out was never less than immediate and was always accompanied by patience far beyond the call of duty.

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An attempt has been made to maintain the scales for stress and temperature contour diagrams uniform throughout this report to allow quick cross-comparison of results. Although there are exceptions, the majority of similar diagrams (i.e. pad pressure, disc temperature etc.) can be compared in this way. This does not apply to graphs.

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Declaration

The work contained in this thesis has not been previously submitted for consideration for a degree. It is entirely the author's own work.

List of Symbols

Throughout this thesis the following symbols are used.

- F — Fourier's Number;
- Pr — Prandtl Number;
- Re — Reynold's Number;
- A_p — pad contact area;
- A_s — braking face swept area;
- Cp — specific heat capacity;
- P — pressure;
- Q — heat energy;
- h — heat transfer coefficient;
- k — thermal conductivity;
- r — radius of disc;
- t — elapsed time since start of braking;
- v — velocity;
- v_{max} — maximum vehicle speed;
- σ — stress;
- ρ — mass density;
- ϵ — element width;
- μ — coefficient of friction;
- γ — thermal partitioning ratio;
- ω — angular velocity of spinning disc;
- $'p'$ — subscript denoting pad;
- $'r'$ — subscript denoting rotor;

Throughout the following report S.I.(mm) units are used in the presentation of results.

CHAPTER 1

INTRODUCTION

1.1 Project Origin and Aims

The project originated from conversations with Peter Gregory of Special Vehicle Engineering¹, Ford Motor Company Limited and was developed for two main reasons. Firstly, brake disc design is not a fully investigated procedure and secondly, the undesirable thermal deformation of brake discs is common in the severe environment in which S.V.E. vehicles operate. This distortion of the disc results in a deterioration of braking efficiency, especially over a number of consecutive aggressive brake applications, and also produces noise or vibration detectable by the driver (unacceptable in a production vehicle).

A programme of research was then devised, undertaken as a BSc. Final Year Project, dealing with the contributions of disc rotor thickness, number of ventilating ribs and the thickness of these ribs to disc distortion. This was achieved by varying isolated dimensions in an attempt to gain information to allow the design of a more mass efficient disc. In practice the only economical way of achieving this was to use a

¹ Special Vehicle Engineering is a specialised department within Ford's Product Development Group which is concerned with the development of high performance variants, intended both for sale and for motorsport competition. Examples of such vehicles include the Sierra RS Cosworth and the Escort RS Turbo.



Finite Element Method computer software package to simulate disc response. An accurate computer model of a 278mm Cosworth Sierra ventilated disc was constructed and used to gain information on the influence of several design features on its performance. Some disc design guidelines were produced at the end of the study although it was accepted that the combination of time limitations imposed by the Final Year Project structure and a lack of Finite Element Method experience meant that the conclusions were not sound enough to justify the investment required to change disc designs, undergo the necessary testing and homologation required, and put the new designs into production.

However, the report did demonstrate the feasibility of this type of study and created sufficient interest within Ford to consider further work which had been shown to be both feasible and desirable. After the preparation of a research proposal and some negotiation, funding for a further full year's research was made available by Ford Chassis Engineering (the group responsible for brakes on all mainstream Ford of Europe vehicles). Finite Element computer simulation work was to be continued at Durham University whilst Ford Research and Engineering Centre in Basildon, Essex would provide technical and some experimental support, accurate dynamometer facilities being available there. At this time the Company was installing thermal imaging equipment for the analysis of transient brake temperatures and it was suggested that work done at Durham might proceed in parallel with experimental testing at Ford.

The research documented in this report commenced in July 1992 and was aimed at making more fundamental design changes to the disc using the experience gained in the previous project. Its ultimate aim was the production of design guidelines, allowing disc design to be carried out with greater ease and confidence through the improved understanding of the reasons and mechanisms of disc thermal deformation.

1.2 Disc Brakes

1.2.1 History

Disc brakes were originally developed in the aircraft industry. They were introduced to the motor industry in the early 1950's on the Jaguar C-type, which used loose round

pads, and then on the D-type, the first car to have discs on all wheels (six circular friction pads acting on each disc!). In the 1960's the disc began to enjoy more widespread utilisation superseding drum brakes (usually on front wheels only - where the braking load is more severe due to the weight transfer), because of its increased efficiency, low weight and improved 'fade' performance in comparison with drums. Brake 'fade' was a considerable problem with drum systems and meant that during heavy braking the vehicle would decelerate less rapidly towards the end of the stop, a dangerous and undesirable phenomenon. Disc brakes have become more and more popular and are currently found on the front wheels of virtually all passenger cars (drums usually being employed on the rear) and on all axles of many high performance vehicles. Many Commercial Vehicles also use discs, although their exposed nature can present problems in certain heavy duty environments. In addition many locomotives and rolling stock utilise disc brakes.

1.2.2 Brake System - function.

The function of any braking system is to retard the vehicle smoothly, consistently and predictably. It must have the capability to slow the vehicle rapidly if required (nowadays 1g is not an uncommon deceleration figure for emergency stop braking) and from high speeds. It must respond quickly when activated and must maintain stability throughout the stop, having no variation or imbalance across the vehicle that might lead to a loss of directional stability and an ensuing loss of control over the vehicle by the driver. It must be able to be operated to full effect by a person of any reasonable age or strength. There are many more requirements with which the braking system must comply, including many legal regulations, but in the context of this report 'smoothly and consistently' are important as they stem directly from the disc/pad/caliper assembly design that is examined here.

The braking components, be they based on disc or drum systems, have their own specific requirements. Primarily a braking torque must be provided on the rotating axle. This is achieved by a friction pair, one part rotating with the wheel (the 'rotor') and the other stationary and indirectly mounted to the vehicle chassis (the 'stator'). The rotor is mounted on the rotating axle that is to be braked and is often located inside the wheel itself. Its primary function is to transmit the mechanical braking torque caused by the very high frictional forces on the spinning rotor to the axle and subsequently the wheels, slowing the vehicle.

The high frictional forces generate large amounts of heat as the kinetic energy of the moving vehicle is converted to heat energy in the friction pair during braking. The secondary function of the rotor/stator pair is to dissipate this heat energy preventing excessive heat accumulation, high temperatures and the resulting damage and deterioration of surrounding components.

1.2.3 Foundation Brake - types

Historically there are two main methods of achieving this friction pair, both of which are still in common usage; drum brakes and disc brakes. Fig 1 shows, schematically, the operation of the two types. Braking with a drum brake is achieved by applying outward pressure with stationary friction pads ('brake shoes' - stators) to the interior of a spinning drum (rotor). The pads oppose one another but drum distortion occurs and can take the form of 'ovalling' and 'barrelling'. Pad wear distribution depends on the actuating mechanism but it is not always uniform. The friction pair is enclosed within the drum which protects it from road debris but does nothing to encourage effective cooling.

In the case of the disc brake two stationary pads (stators) oppose each other across the thickness of the spinning disc (rotor). The disc is inherently much more exposed and better convective cooling is achieved even though the design is prone to the invasion of debris (dust shields can be fitted). In addition there is a shorter thermal path for heat to be conducted from the braking faces to the axle/wheel.

1.2.4 Brake Disc - overview

There are two major types of car brake disc; solid and ventilated. The ventilated disc gives increased cooling ability without a proportional increase in weight, but it is more complex in its design and is not always necessary. In the case of high performance cars where a ventilated disc is required to achieve the desired heat transfer, the disc is commonly constructed of a cast iron hub and two annular braking faces, separated by radial ribs (also known as vanes). Air is able to flow through the air passages between the ribs, carrying heat away from the braking surfaces. A solid disc, as its name suggests, has no such air passages and as such has poorer cooling performance. Fig 2 shows a ventilated brake disc, its section and terminology. This thesis is concerned with the design of ventilated brake discs.

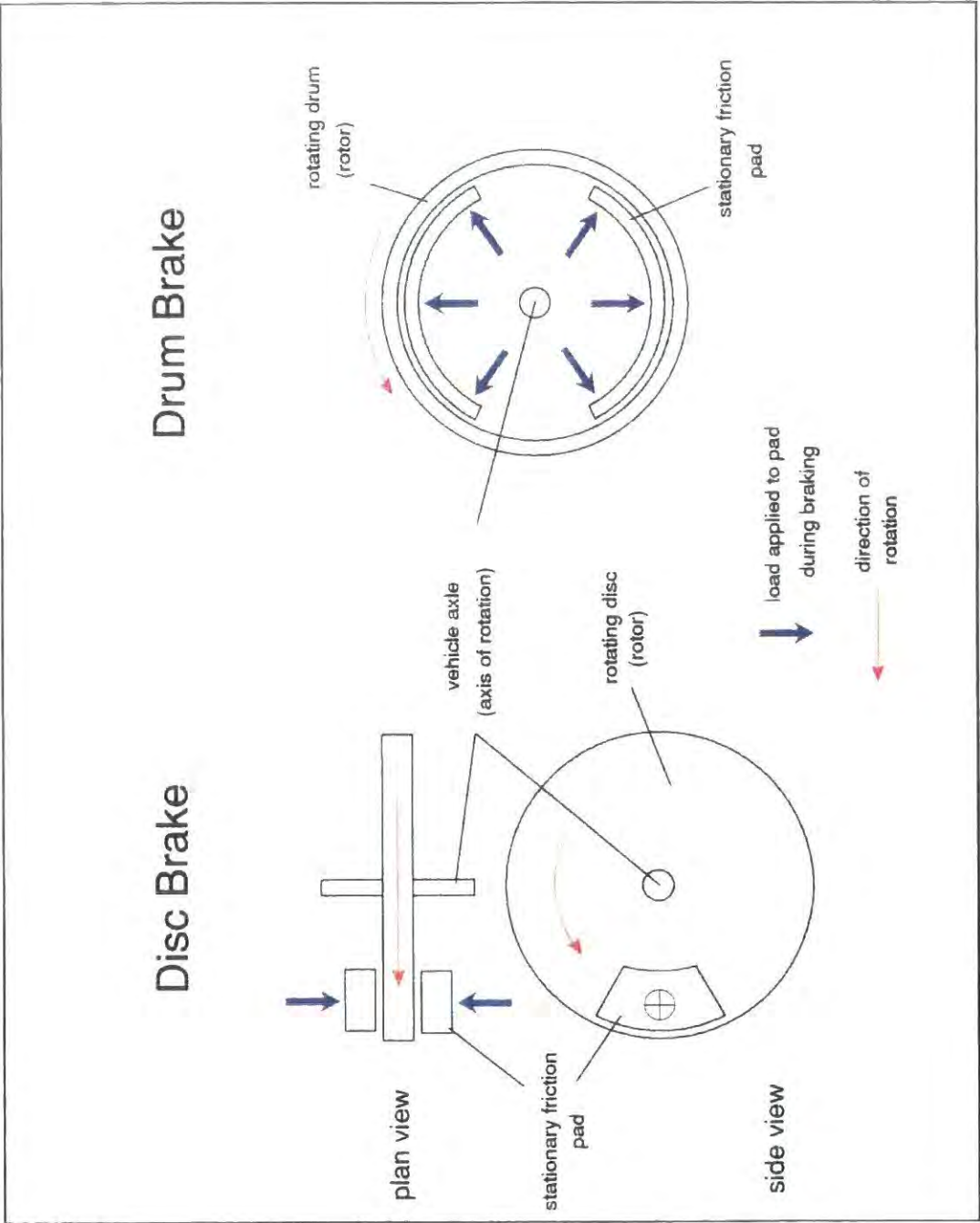


Fig 1. Schematic diagram of drum and disc brakes.

Conventional Ventilated Brake Disc – section removed

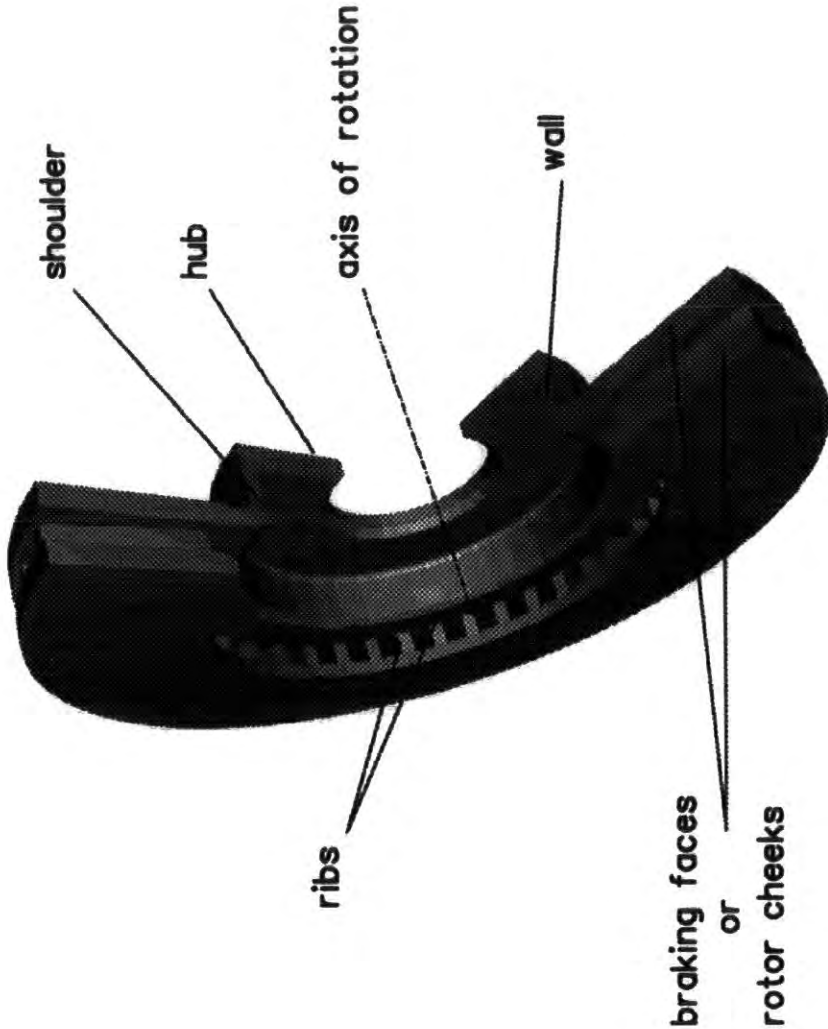


Fig 2. Ventilated brake disc - terminology.

1.2.5 Ventilated Brake Discs - types

There are two types of ventilated disc with which Ford is concerned; conventional and 'inside-out'. The 'inside-out' design was first described by Inoue [18] and the difference between the two lies in the disc cross-section between the hub and the braking faces. In these discs one face can be thought of as being directly attached to the hub via a continuous section of material whereas the other is supported solely on the ventilating ribs. In a conventional design this 'supported' face is on the inside of the vehicle as it mounted on the axle and on the 'inside-out' version this is reversed and the supported face is on the outer side of the disc. This is illustrated in Fig 3, the difference in cross-section being highlighted in the diagram.

1.2.6 Disc Brake Caliper Assembly

The caliper/pad forms the stator function of the friction pair and is the actuating mechanism for applying the reaction force for the brake pads. It is usually a solid casting containing a hydraulic system to provide the force on the back of the disc pads. In this case due to constraints of mounting space inside the wheel the caliper contains only one hydraulic cylinder and this is used to create a reaction force between the pad it directly acts upon and an opposite pad restrained by the opposite side of the caliper. This caliper is required to be mounted on a sliding pin arrangement so that a small amount of lateral movement is available to create equal and opposing forces across the thickness of the disc. Fig 4 illustrates the caliper in simple form. It is also common to have a rigidly mounted caliper with two opposing hydraulic cylinders to provide balanced pad forces. The disc pads are removable and need to be replaced periodically due to wear. The pad itself consists of a steel backplate onto which is mounted (usually using very specialist adhesives) a softer friction pad. This pad, a fibre composite, used to be a combination of asbestos and a binding resin but more modern purpose-made materials have been developed with specially engineered properties. The pad is forced into contact with the spinning disc and is subjected to very high pressures, temperatures and compressive and shear loads.

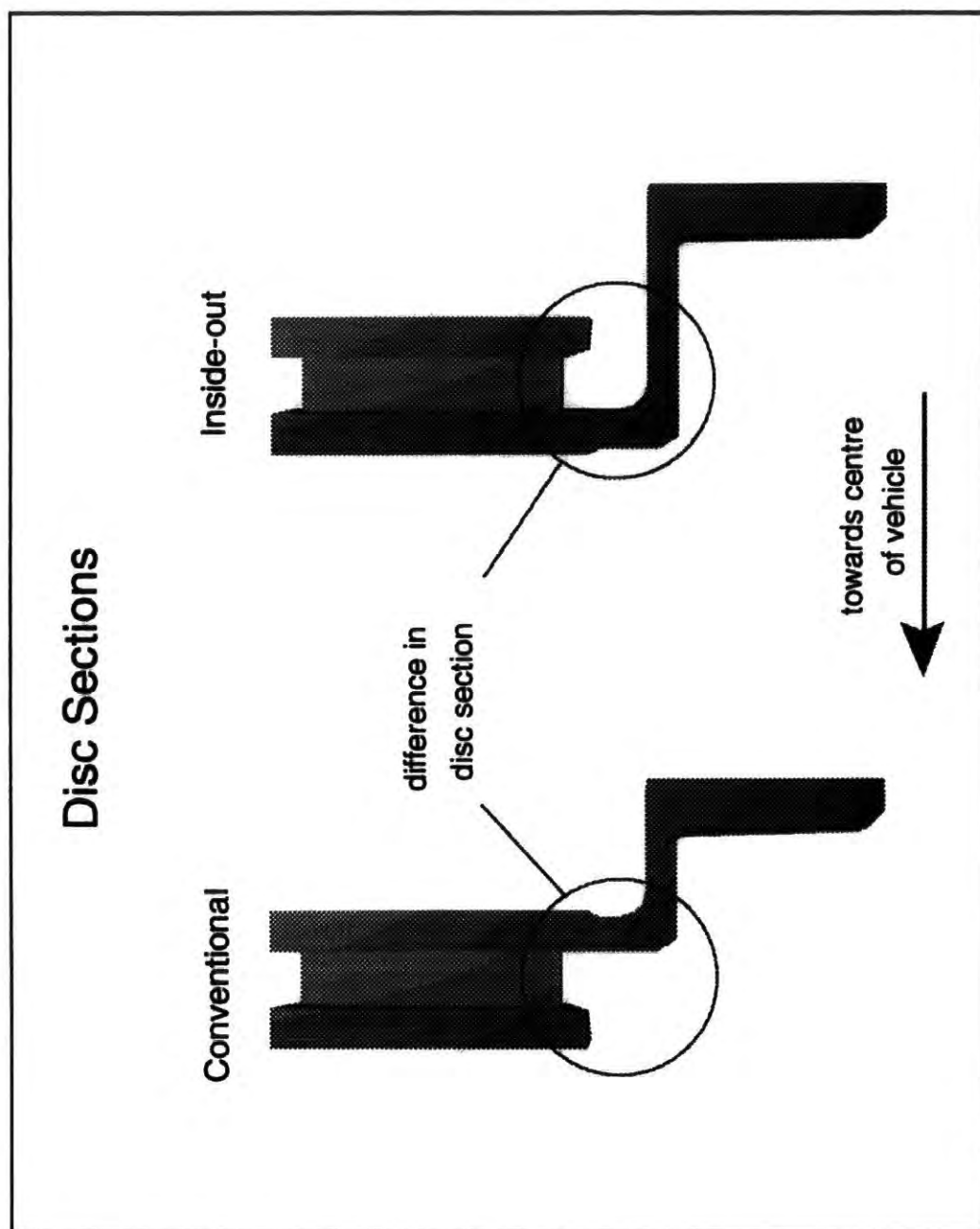


Fig 3. Conventional and 'inside-out' disc sections.

Caliper/Disc Section

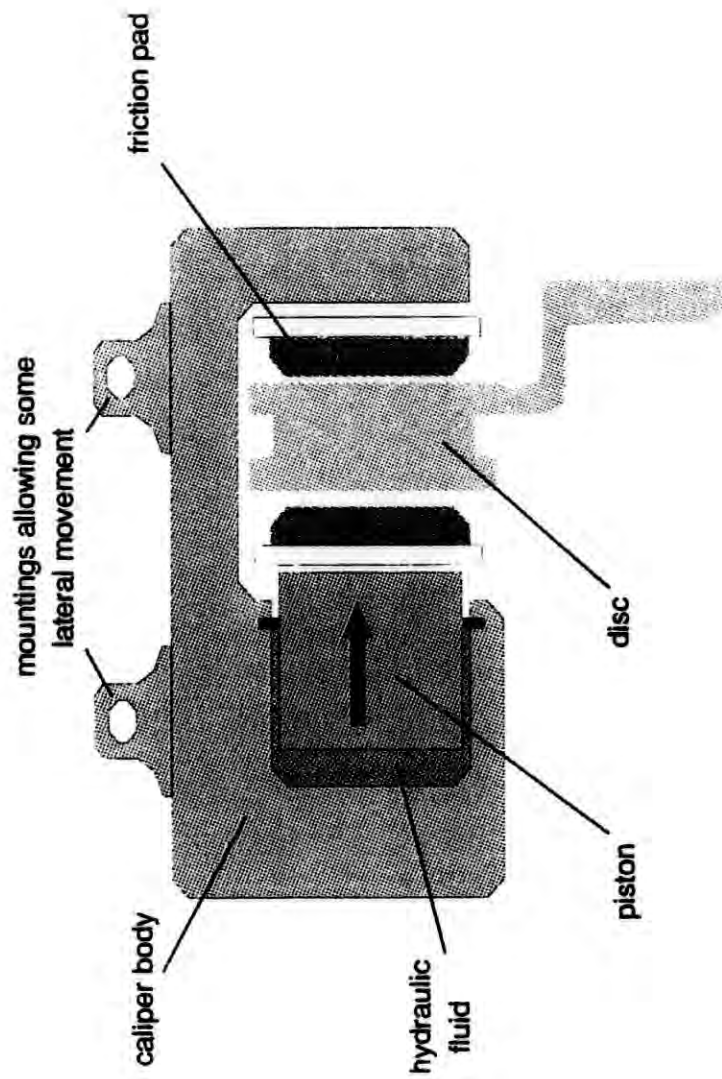


Fig 4. Schematic diagram of caliper.

1.3 Brake Characteristics and Limitations

A brief explanation of both the desirable qualities and the problems encountered in service is required.

1.3.1 Disc

Brake discs are subject to three main problems in service; cracking, wear and distortion. Cracking occurs usually because of high stresses (often cyclic in nature) caused by temperature differences across parts of the disc. These cracks often appear in the braking face where temperatures are highest and result in severe weaknesses in the disc and increased pad face wear (these cracks can grow rapidly because of the cyclic nature of brake disc operation). It is important to select both disc design and material carefully to ensure that any thermal deformation that takes place is not so great as to put an inordinate amount of stress on the component. Therefore if brake disc distortion can be reduced it is likely that corresponding stresses and therefore potential cracking will be reduced. Disc wear is caused by the rubbing of the pad on the rotating disc and is described in more detail in Section 1.3.2.

There are two types of distortion that can occur during brake operation. The first is due to the high mechanical forces on the disc during heavy braking and the transmission of the braking torque to the axle. The disc must be stiff enough to withstand these loads without raising internal stresses to a level where they either generate cracks or enter a plastic deformation region. The disc must be designed to remain within its elastic zone at all instances of operation, maintaining dimensional stability.

The second, and main, form of disc distortion is thermal deformation arising from large temperature differences across the disc due to heavy braking. Four basic types of thermal deformation occur in discs, and the three most significant are illustrated in Fig 5.

- i) **radial expansion** - produces a radial displacement of the braking face at the position of the brake pads on the surface of the disc.

- ii) **coning** - results in an axial deflection at the outer edge of the braking faces producing a conical deformation of the disc. This forces the pads apart slightly, changing the pressure distribution of the pad on the disc and hence creating an uneven heat input to the braking face. Braking effectiveness will be reduced as a result.
- iii) **waving** - results in torque oscillations as the wheel turns, introducing instability to the braking system and a loss of smooth operation. This torque fluctuation also manifests itself in 'brake judder', a low frequency noise and vibration transmitted to the driver through the suspension, chassis and steering gear of the vehicle. This effect is easily detectable by the driver and is unacceptable in a modern production car. However its causes are not fully understood and its occurrence is of concern to Ford.
- iv) **rippling** - takes the form of localised deformation between the ventilation ribs due to the thermal stresses, and resulting strains, induced between the constraints of the ribs around the rotor braking face. This adversely affects pad contact pressure distribution and stability resulting in reduced braking efficiency and possibly in such effects as 'squeal' where a high frequency vibration of the pad is initiated causing loud, high-pitched noise during certain braking conditions.

Of these deformation characteristics, waving and coning are the most important having the largest effect on braking performance and hence the majority of this work is concerned with the study of these phenomena in an attempt to isolate the reasons for their occurrence and the mechanism by which they evolve.

It is important that the disc must be designed to conduct heat to the hub effectively and without incurring large distortions such as those described above. It must also be able to lose heat to its surroundings by forced convection and it is very important that the material and design is able to withstand the continual thermal cycling without loss of performance or integrity.

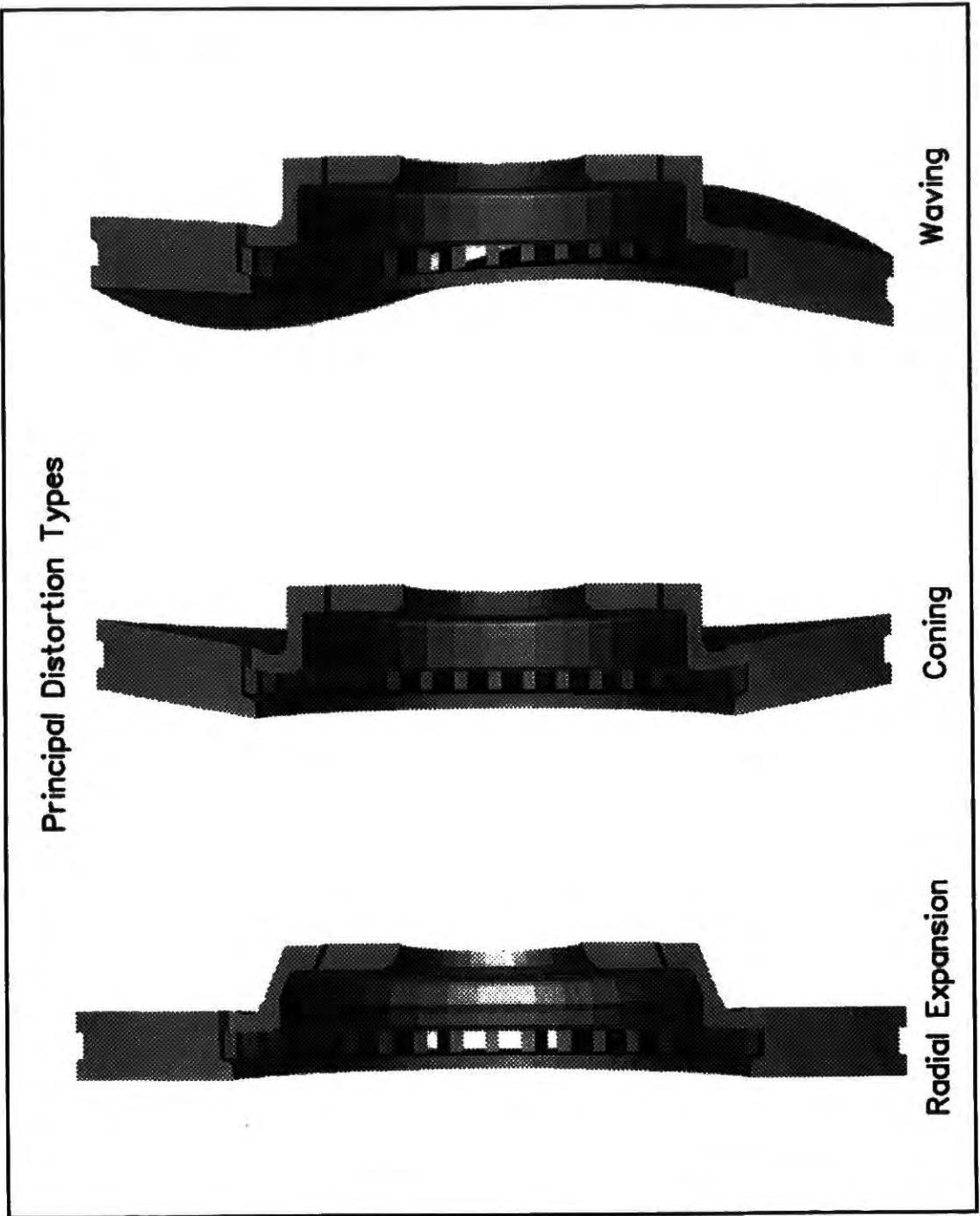


Fig 5. Forms of thermal deformation.

1.3.2 Pad

If the pad is poorly designed it will distort under heavy loading resulting in uneven pressure distribution and even in some severe cases incomplete contact of the pad with the disc. The resulting uneven heat input into the braking face can result in localised heat build up with associated localised high temperature regions. The pad backplate stiffness must be carefully determined along with the optimum friction pad material compressibility to give sufficient pad rigidity for caliper piston loads to be transmitted evenly and uniformly over disc/pad contact area.

As very high temperatures are developed at the interface it is also vital that the pad friction material has a very low thermal conductivity. Any significant increase in caliper bulk temperature would result in brake fluid degradation and impaired function of the braking system. In addition rubber piston seals would be damaged, considerably reducing the operating life of the components.

The wear characteristic of the friction material must be carefully selected to ensure that the proposed pressures, surface speed and temperatures will not cause the material to wear too quickly (material too soft) or to cause too much wear to the disc in the off-brake position (material too abrasive). A friction film is developed in the high temperature/pressure contact region and this film adheres to the disc surface. This has only recently been studied and research is emerging on the characteristics of this film and its importance.

If the pad material is too abrasive then disc wear will take place during off-brake operation¹. If significant disc run-out is evident and the pad material abrasive then this wear will be localised, causing judder during a brake application that is unrelated to thermal distortion (cold judder caused by 'disc thickness variation' [28]).

¹ Even when the brakes are not being applied there is a continual rubbing contact between the pad and the disc. This is principally due to the piston seal design and frictions in the hydraulic system and the sliding caliper mounting. Some contact is desirable for the pad to perform a 'cleaning' function on the braking faces of the disc but excessive off-brake contact forces will result in disc wear.

Pad material compressive properties are important in determining pad pressure distributions especially in the presence of either disc or caliper distortion. Although this work is directed at disc design, pad and caliper design are also vitally important and further mention of pad properties is made later.

1.3.3 Caliper

The most important attribute of the caliper is stiffness. Any flexure in the component results in it 'opening out' under loading (illustrated in Fig 6.). This inevitably causes uneven loading of the pad backplate and hence uneven pressure distribution at the disc/pad interface. In extreme cases contact and binding can occur between the piston and cylinder bore with resulting wear and uneven piston operation. As an unsprung component it is also desirable for the caliper to be as light as possible without adversely affecting its stiffness. Some capability for heat dissipation must be included in its design.

1.4 Detailed Project Brief

The design of the brake disc is becoming more and more critical as modern discs are subjected to more loading. The disc is being buried in deeper and deeper wheels, necessary for high performance vehicles (and desired vehicle image) and the increase of aerodynamic efficiency is also causing a conflict of interest [15] in modern cars. As air is directed away from the underside of the vehicle more efficiently with air dams and front spoiler arrangements, less cooling air is directed over the disc. It is not uncommon for motorsport vehicles to cause their discs to glow with heat accumulation after a small number of aggressive stops from high speed and this temperature rise is harmful to surrounding components. Braking efficiency is lost through increased wear and changing frictional characteristics. High temperatures can also cause the brake fluid to boil and the piston seals within the calipers to be damaged. In addition, the resulting deformation of the disc reduces even pad contact and stopping ability.

Exaggerated Caliper Deflection

caliper flexure increases contact
pressure at outer edge
of pad #1

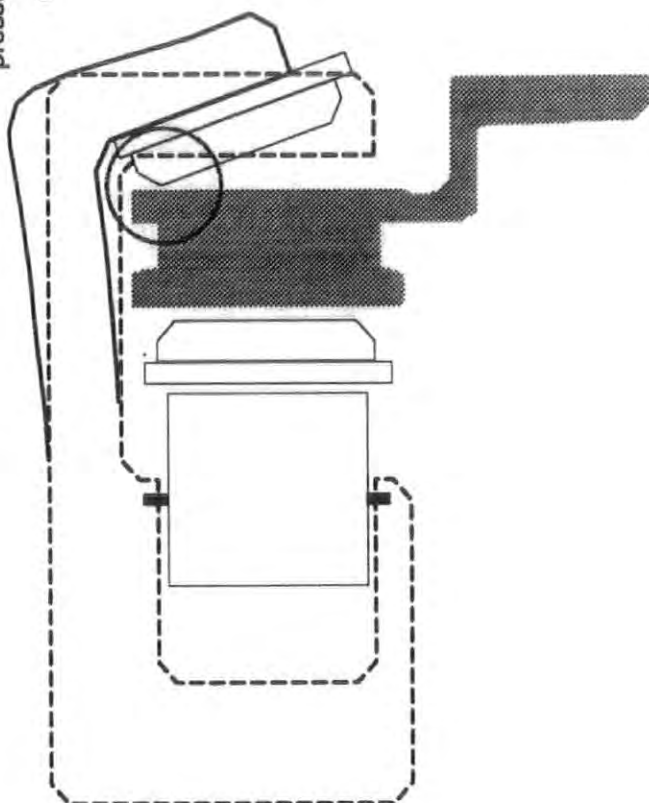


Fig 6. Caliper flexure.

It is evident that the brake disc must become more and more efficient and although other developments are possible, for example changes to wheel/cover design to introduce cooling air through the wheel to the disc, helping to maintain brake disc efficiency, improvements in the design of the disc itself must be made. In all research encountered, few established principles or relationships governing the different variables affecting brake disc design are available to aid the engineer in obtaining this increased performance.

This research was founded primarily to examine the causes and mechanisms of the thermal deformations described above and in doing so to try and develop improvements in design based on proven disc behaviour. Integral to this is an examination of the relative merits of conventional and 'inside-out' discs.

Conventional and 'inside-out' disc types have been seen to have different distortion characteristics and cooling properties and although to some extent it may vary from one disc design to another of the same type, the reasons for their behaviour have not been fully investigated and the differences understood and qualified. It was a priority of Ford in funding this work that an unbiased and independent investigation into the merits of each was made (much F.E. work is done for the Company by brake suppliers, often in support of their own product and so is not always felt to be entirely independent or objective). Both types of disc are currently used on Ford vehicles but their selection is not based on established principles but on an element of empirical observation. Some aid in the selection of which type of disc is suitable for a given situation is sought. In addition independent confirmation of some previous work by suppliers was also sought.

The final aim of the project was to utilise the experience gained in examining the distortion mechanism to produce detailed guidelines, aiding brake disc designers to extract maximum performance from a disc under specified conditions, maintaining the disc mass as low as possible. It was recognised that there would be no 'hard and fast' set of rules that would apply in every case of disc design but that there would probably be preferred directions to pursue according to the intended operating conditions of the disc. For example, a disc may be subjected to continual low speed, check braking with few heavy applications (town driving) where a lighter, faster cooling disc would be utilised. Alternatively, a disc intended for faster driving (such as consistent motorway use) is likely to have infrequent but heavy applications and this type of disc

would have a large thermal capacity (mass) to cope with the single demanding applications (energy required to be absorbed by the braking system increases with the square of the vehicle speed) but having longer to cool in between applications. Therefore a detailed description of the mechanism of brake disc distortion followed by a series of analyses and recommendations on the effect of certain disc features/characteristics would be of some value to the engineer. An illustration of an improved design, devised according to a given set of desirable characteristics, would be given along with the details of its relative performance.

Reduction in thermal deformation was the primary measure of an improved design but absolute disc temperatures and internal stresses (cause of cracking) were also carefully examined in all trials.

Disc mass is central to disc design, it fundamentally affects thermal performance as a greater mass means a greater capacity to absorb heat energy ('thermal capacity'). However, the reduction of weight of a brake disc without losing its performance is very desirable, not only to reduce material costs (even a small reduction in piece cost is worth achieving in a mass production environment as the saving is multiplied many times throughout a model's production life) but particularly because of the benefit to vehicle suspension performance. A reduction of unsprung weight improves the response of the spring/damper system to sudden shocks.

Throughout, the disc mass was borne in mind as a part of the measure of a successful design of disc (low deformation combined with low mass meant an 'efficient' design) but it was not a fundamental concern as in the previous project. However it is important in disc design to differentiate between those increases in disc performance attributable solely to an increase in disc mass, and therefore thermal capacity, and those that are down to an improved design of the fundamental structure of the disc.

The study and improvement of component materials or cold judder effects was not included in the project brief.

1.4.1 Brake Packaging - strategy

Before commencing it was necessary to define a strategy for design changes. Wild and outlandish changes to the current design would require vast re-engineering of the

suspension/wheel/axle assembly and this is clearly unjustifiable and undesirable. It was therefore specified by Ford that any design changes should be made approximately within the existing envelope i.e. the disc diameter should not exceed the current value to fit the current wheel size/caliper location. Disc position is quite strictly governed by suspension geometry, wheel dimensions and the packaging and location of the disc itself within the design of caliper. However Ford was also interested in recommendations for future vehicle designs where more flexibility would be available when the vehicle was being designed from the early stages.

Cost and ease of manufacture was considered at all stages in the creation of revised designs of disc.

1.4.2 Research Programme Outline

The research was to be founded on an 'inside-out', ventilated disc, utilised at the time of the research project on the more high powered derivatives of the CDW27 - Ford 'Mondeo'. Details of the disc, caliper and pads used and referred to throughout this report are given in Appendix I. The disc itself was a $\phi 278\text{mm}$, grey cast iron disc and the pads and caliper were of the type designed and manufactured by Alfred Teves GmbH.

Five main phases of the work were outlined

1. Development of an accurate Finite Element mesh.
2. Application of simulated mechanical and thermal loading conditions.
3. Validations of the disc models.
4. Detailed experimental work using models.
5. Proposal of improved design based on experimental results.

The first and second phases are described in Section 3, with the remainder being detailed in Sections 4,5 and 6 respectively.

CHAPTER 2

LITERATURE RESEARCH AND REVIEW

2.1 Information Sources

The original research undertaken for the previous project was greatly expanded by examination of wider subjects, other database listings and by more rigorous scrutiny of references. The following sources were utilised.

1. Durham University Library - A manual search of periodicals and proceedings revealed many relevant papers. Proceedings of the Institution of Mechanical Engineers, 'Strain', 'Journal of Mechanical Engineering Science' and others were available. In addition, the Current Technology Index (formerly the British Technology Index, and a subject listing of technical articles published within a particular year) proved invaluable in providing references for papers. In this way papers going back to 1958 were collected, practically all the I.Mech.E. issues being obtained from the library stock.
2. Ford Motor Company Technical Library - I was fortunate to be given access to this very well stocked library covering virtually all publications within the motor industry, both in Great Britain and the United States. In addition to I.Mech.E. journals there was also a regularly updated catalogue of Society of Automotive Engineers publications of which several proved relevant. Database searches were also conducted through Ford's on-line

system revealing several more papers, all stocked by the library. Other titles such as the 'International Journal of Vehicle Design' and 'Engineering' were available.

3. Bath Information & Data Service (B.I.D.S) - Some references were also obtained by manual on-line searching of the Science Citation Index maintained through the B.I.D.S. connection from University computers.

Over 30 scientific papers relating to brake design were obtained, ranging from 1958 to 1993, and mainly being published under S.A.E. or I.Mech.E. titles. These were invaluable in ascertaining the current level of work in this field and in providing some theoretical and experimental proof of certain concepts that owing to time limitations could not have been completed at Durham.

2.2 Chronological Review

T.P. Newcomb is rightly regarded as one of the pioneers of research, both theoretical and experimental, into the behaviour of discrete brake components. Whilst a Research Officer at Ferodo, he published four papers [1,2,3,4] between 1958 and 1960, all concerned with the development of transient temperatures and employing Laplace transformation techniques in producing analytical solutions to various simplified problems. He produced solutions for drums, discs and dry friction clutch transmissions operating under various conditions and also suggested methods for sizing components, a vital step in producing research of significant value to the brake design engineer. Although the solutions contained, by modern standards, unacceptable approximations and were often based on infinite or semi-infinite slab theory the proposed results were important in the development of analytical theory in the field and have been repeatedly referenced since.

In 1965, while still at Ferodo, Newcomb joined with Millner [5] to produce an experimental account of disc and drum cooling. Drag tests were performed on vehicles and then cooling rates of the components were obtained with thermocouples. They produced an empirically derived formula relating cooling to vehicle speed which accounted for different sizes of drums/discs. However this equation could not account for the differences in cooling air flow over the discs caused by different vehicle body

designs. Ventilated discs were compared to solid and the effects of fitted dust shields were examined. It was estimated that cooling rates for front brakes are 20% higher than for rear brakes and that disc brakes cool 25% faster than drums of an equivalent capacity. Dust shields reduced cooling by upto 30%.

Abbas, Cubitt and Hooke [6] highlighted the distortion effect of coning and using a computer program, produced a method of calculating stresses in a solid brake disc from an initial known temperature distribution. This method, based on finite numerical methods, could be used to produce a disc less susceptible to this form of deformation by careful design of its section. An illustration is given of the method being employed to design an improved disc. Abbas [7] followed this with a further paper, relating experimental work to confirm the previous theoretical research, using a static test rig to heat the disc and then allow it to cool, whilst thermocouples and strain gauges recorded the data. Good agreement was found.

El-Sherbiny and Newcomb [8] derived a numerical method in 1976, following Abbas' lead using finite difference techniques to analyse temperature distributions in dry clutches under a variety of different conditions. Numerical methods were now being used because the increased complexity of problems and original assumptions that had now been found to be unacceptable could not be contained within a purely analytical solution. Joining Ashworth, El-Sherbiny and Newcomb [9] proceeded to apply their method to brake drums. In both papers such complex situations as incomplete contact between the friction pair were examined, even though they only had access to limited computing power and a relatively simple Finite Element method. Distortion was able to be modelled for a variety of contact cases and modifications to the drum made for improved cooling and therefore thermal distortion.

Sisson [11] uses Duhamel's theorem to combine the advantages of the precision of an analytical solution with the approximate solutions of a more geometrically precise model obtained with numerical methods. Using a computer Sisson computed temperature and, unusually, stress distributions theoretically for ventilated brake discs. He then conducted practical experiments to derive empirical relationships for disc vent cooling characteristics to be used in the computer solutions (these relationships were subsequently used by Sheridan, Kutchev and Samie) and are the only documented relationships for vent airflow available. The complexity of vent air flow is such that it is very difficult to measure, even harder to model and is almost uniformly neglected in

theoretical calculations of ventilated disc performance. Sisson concludes by demonstrating the accuracy of his theoretical solutions in comparison with dynamometer test results.

Disc pad flexure and its effect on pressure/temperature generation in the pad/disc interface was considered by Harding and Wintle [12] in 1978. They started by likening the physical system of the pad under load to a much more simplified arrangement. Experiments were then conducted in conjunction with analytical theory and finally a finite element model was produced. Important work was completed relating pad material properties to pedal travel effects, pad pressure distributions and pad distortion. Studies on the effect of pad backplate thickness and friction material compressibility were undertaken. They conclude that a combination of high backplate flexural stiffness and high friction material compressibility is desirable, not only for uniform pressure distributions but to prevent a loss of contact between the disc and pad at the leading and trailing edges of the pad. Static and dynamic loading was applied to the pads.

Day, Harding and Newcomb [13] produced a comprehensive modelling system using the PAFEC finite element software package in 1979. Until this point brake analysis methods had ignored the effects of shoe and drum distortion on contact pressure distribution and an iterative system was proposed to model the friction interface. A two dimensional model was developed that allowed the brake drum and shoe to distort during the simulation, the varying contact area and resulting pressure distribution being calculated. This was an important step in the modelling of this dynamic system and arose primarily because of the increase of available computing power and software, allowing a complex, iterative solution to be obtained. Component distortions and torque fluctuations were calculated and the predictions verified with a programme of experimental work, the conclusion being drawn that a much more accurate analysis could be obtained by using an iterative solution scheme. There were limitations to the accuracy in that thermal distortion was not being accounted for and the finite element model of the drum was crude.

At about the same time Timtner [14] demonstrated a scheme for the analysis of brake components using computer-based finite element analysis although not of the complexity described above and in 1984 Day and Newcomb [16] returned with further work achieved using an extension of the computer scheme outlined in the paper above.

In this exposition frictional heat generated during braking and wear on the components were included in the analysis, whereas before only the mechanical forces and displacements had been examined. Drum brakes were again analysed, the simulation stepping through a single stop, at each point calculating distortion due to mechanical and thermal stresses and revising the pad contact pressure distribution and therefore the heat input to the drum. A more complex and more accurate model was utilised than in the previous work, the whole drum being modelled in two dimensions. A further paper by the same authors [17] in the same year applied a similar system to the friction interface of a disc brake. A relatively simply model was devised, an axisymmetric (two dimensional cross-section) finite element mesh of the disc and the pad, and the heat exchange between the two sliding bodies was examined. This study again used the iterative scheme and incorporated wear and thermal distortion in its computations. Experimental work again confirmed results obtained by calculation.

Inoue of the Toyota Motor Corporation published an important paper in 1986 [18] combining disc rotor transient thermal measurements and finite element work. He started by devising a system for measurement of ventilated disc temperatures during a stop, recording them for later display and analysis. Several interesting phenomena were documented. Firstly after a single stop on a dynamometer some plastic deformation was found to occur in the form of a double wave around the circumference of the brake disc. A corresponding torque fluctuation was measured. In addition 'hot spots' were encountered on the braking faces of the disc. Hot rings had previously been observed but when the ability to view the temperatures frozen on the disc during part of a braking application was available then localised high temperatures were found. These hot spots did not remain stationary and grew and disappeared at apparently arbitrary positions, although in a cyclic manner, over a period of many brake applications. The measurements did show that not only were the temperatures not uniform in a circumferential direction around the disc but they were also uneven in a radial direction over the disc surface. The first assumption here was universal among previous papers predicting disc behaviour and it was usually assumed that the contact pressure had no initial variation with radial position. Inoue then went on to create a finite element model, subjecting it to the results of the dynamometer tests as thermal input. Again a waveform deformation was observed in the results of the simulation. In addition Inoue related initial disc runout to peaks of temperature on the disc braking face. The paper concluded by proposing a new form of disc, the 'inside-out' configuration described in the previous Section, specially

designed to combat coning and reduce brake judder. Its ability was demonstrated in further computer simulations.

Sheridan, Kutchey and Samie [19] produced an extensive work in 1988 providing several detailed guidelines for brake design engineers. Four separate models were produced, ranging from a one dimensional model for rotor bulk temperatures, through a two dimensional steady-state model for predicting plateau temperatures during a multi-stop braking schedule and finally to a three dimensional transient model for the simulation of surface temperatures during a stop. All were found to be useful depending on the stage of disc design. However none of the models incorporated variation of heat input either over the pad contact area or with time which is a major source of error. Good correlations are demonstrated with actual vehicle measurements and the paper also contains detailed theoretical equations for heat transfer coefficients at different locations on the disc. Some are entirely theoretically derived and others are empirical, Sisson's vent cooling equations making a contribution.

Missori and Sili [20] produced work on rolling stock brake discs, creating a very simple finite element model and varying specific dimensions to gauge their effect on disc thermal response. Different operating conditions were simulated. In the same year Ramachandra and Ramasubramanian [21] produced a finite element model of a commercial vehicle drum for fade analysis, examining the effect of changing contact area throughout a single brake application.

Watson, with Newcomb, [23] presented a paper in 1990 detailing a three dimensional method of analysis for drum brakes. This again was a further extension of the system developed by Day at Loughborough University in the early 1980's and now drum distortion in both axial and radial directions were encompassed in the study. Variations in shoe pressure distribution varied during a single stop simulation, the iterative scheme again being utilised, and shoe wear and torque fluctuations were predicted. The work concluded that frictional drag per unit area remained reasonably constant over the interface of the drum and the lining and theoretical results were confirmed by dynamometer testing.

Day, Tirovic and Newcomb [24] produced a paper the following year that concentrated on the relationship between localised heat generation at the sliding interface and pressure variation at the same place. Some of the problems, both

microscopic and macroscopic of excessive heat build up are discussed and after observing that many problems associated with brake performance (fade, speed sensitivity etc.) are due to localised heating they concluded that maintaining as near uniform pressure distribution over the face as possible is important in the generation of stable, uniform temperatures in order to avoid poor disc performance and material degradation.

Two further papers by Day and Tirovic [25,26] published in the same year establish more detailed work in the field of interface pressure distributions, one in drum brakes and the other in disc brakes. A more holistic approach is maintained and detailed pad/actuating piston and caliper models are produced in addition to a three dimensional disc model. Caliper flexure is examined and detailed work is produced on the pressure distributions caused by the flexure and distortion of all the components in the system, not just the disc. Both papers conclude that pad and caliper stiffness are crucial but that disc flexure does not effect pressure distributions at the pad face. It recommends further work on the relationship between pressure distributions and thermal effects.

A small number of the papers presented at the 1993 'Braking of Road Vehicles' I.Mech.E. conference, attended by the author, have relevance to the field of brake analysis. Kao, Richmond and Moore [27] presented a paper describing the detailed testing of brake pads, both experimentally and using computer simulations. The accurate modelling of the anisotropic friction material properties constituted an important part of this work. Haigh, Smales and Abe [28] gave an account of their work on cold judder, resulting from uneven disc wear in the off-brake condition. This disc wear was stemming from excessive disc runout and friction pads being too abrasive. This paper highlighted the importance of selection of the correct properties of friction material and concludes with recommendations for improvement of caliper designs and the suggestion for further work in the area of friction material formulation. Methods for appraising vehicles with cold judder and quantifying the severity of the case are given.

An important paper by Börjesson, Eriksson et al., [29] described the role of friction films in automotive brakes. It is now known that a layer develops in the high pressure/temperature interface between the pad and disc and that this layer adheres to the disc and either greatly affects or even governs the frictional properties of the

system. Study of this film is beginning to be initiated in several research centres and the subject is likely to have a greater impact in future years. It is not as yet possible to model this film and work is being conducted within the field of experimental tribology.

2.3 Literature Summary

The extent to which predictive analysis has been able to be carried out in recent years is a reflection of the increased power of computers that has been available. Gradually more and more of the factors that had been forced to become assumptions of uniform behaviour have been examined and included in the modelling process. In the most recent work, wear, distortions during the brake application due to mechanical and thermal stresses and variable material properties have all been modelled in the course of a brake simulation. Throughout the last thirty years there has been a progression in the accuracy available and whilst early analytical techniques have in no way lost all relevance to modern design the selection of those factors that may remain uniform or constant is still vital to a successful piece of research. Inoue's work is of great importance. Not only did he apply the results of accurate experimental work to a finite element model, thereby removing many assumptions about heat input (probably the greatest source of inaccuracy in the majority of the analyses presented here), but he made a very important link between pressure variations caused by residual plastic deformation or runout, and localised heat variations (hot spots). This does not appear to have been fully embraced by subsequent researchers in this field, but is of great significance to the study of the mechanism of disc distortions. Recent work by Day and Newcomb have highlighted the importance of determining the disc/pad pressure distribution as part of the disc modelling process. Uniform pressure and temperatures, whilst highly desirable, can no longer be considered valid assumptions in the modelling of a disc/pad system.

Many of these papers will be referred to again in detail later in this report and the information contained within them formed the vast majority of the research base. 'Braking of Road Vehicles' by Spurr and Newcomb [30] also provided some detail on the mechanics of braking vehicles.

2.4 Historical Limitations of Accuracy

Accuracy of any predictive analysis system is its chief limitation and in the vast majority, the formulation of valid assumptions to enable calculations or modelling to be performed is the chief factor affecting accuracy. Indeed, it is pointless producing analytical or numerical method solutions to any projected degree of accuracy (or even at all) if there is little comprehension of the validity of the assumptions and the effect that they have on the system being modelled. Gradually as more computing power has been made available many of the early assumptions have been investigated and their effects quantified.

It would be unfeasible to chronicle all sources of error made during the papers described above but several important ones stand out.

Only recently has the importance of pad pressure distribution on heat input to the disc been highlighted. Any future work must take pad behaviour into account and, to a lesser extent, caliper flexure.

It has recently been proposed that friction films become a considerable source of error when closely modelling the friction interface conditions but do not have a great effect on general disc modelling and sizing.

Little work has yet been documented using three-dimensional disc models. A two-dimensional model does not allow the calculation of effects of pad load on the disc and variation of heat input around the disc (Inoue showed this to be important - possible causes being disc runout and residual plastic deformation). It is not possible to examine waving distortion in two-dimensional models and this is the primary cause of brake judder.

Disc cooling is seldom examined, even more rarely with any accuracy. This is because it presents difficulties and whilst it is recognised as a major factor in the thermal response of brake discs there exist only simple theoretical and empirical equations representing its effect. The main concern is with air flow over the disc. It naturally greatly affects cooling performance and the difficulty lies in determining exactly what form the cooling air flow takes on the vehicle. This air flow is affected

by where the disc is mounted in the wheel, wheel cover/wheel design, vehicle aerodynamics, underbody and suspension design, vehicle speed and wind direction. This means that it varies from vehicle to vehicle and no adequate assumptions concerning uniform behaviour can exist. In addition disc location means that it is very difficult to place instruments to accurately measure air flow, and even if it were straightforward then the air flow would probably be affected by their presence anyway (measuring disc air passage flow velocity on the vehicle is virtually out of the question). In many of these papers dynamometers are used as an alternative to vehicles (for simplicity, convenience and repeatability of measurement) and often a fan blowing air over the surface of the disc is used to simulate disc air flow. This naturally does not take into account the disturbance of surrounding chassis components, often including a dust shield and wheel cover, and unfortunately accurate dynamometer tests would require many surrounding components and a wind tunnel to create similar conditions to those experienced by the disc on the vehicle. It is therefore not a valid reproduction of a varying, unpredictable and complex system. In addition, although Sisson [11] produced empirical relationships for disc vent cooling flow these were only valid for conventional discs and no such data has been found for the substantially different flow characteristics of 'inside-out' discs. Any cooling studies must be carefully constructed and serious thought given to any results based on assumed behaviour.

The strength of finite element modelling lies in its capacity to perform comparative tests. Producing the results of a single test and proclaiming with any degree of confidence that this is the absolute result that would be obtained in service would be foolish in the case of brake analysis. The depth of information required to do this successfully would prove prohibitive if it were intended to produce such results. However, if a disc and its environment is modelled with reasonable accuracy then a change to the disc is likely to yield a proportional change to the predicted value consistent with the equivalent change made to a real disc in service.

It is apparent that any modelling requires careful consideration of assumptions made, the effect that these assumptions have on the result and the extent to which the real conditions are to be modelled. These are discussed in relation to this work in the next Section.

CHAPTER 3

COMPUTATIONAL METHOD

3.1 Overview

3.1.1 Available Resources

Computing facilities at the University centre on a UNIX system, a network primarily containing HEWLETT-PACKARD workstations but also serviced by IBM PC connections. As had been the case with the original project all finite element calculations were to be performed on the PAFEC75 software package (described in the following Section) with some post-processing of results to be performed through UNIRAS graphics software. UNIRAS is the collective name given to a suite of programs, some dedicated to data mapping, graphing and image processing in addition to a library of graphics subroutines available for use in users' own programs. This graphics library was utilised extensively and is described further in Section 3.1.3. Many PC's were available and spreadsheets were used for both analysis and presentation of results.

3.1.2 PAFEC75 - description

PAFEC is an acronym for 'Program for Automatic Finite Element Calculations'. It was first developed in the 1960's at Nottingham University but has been continually developed and extended and is now very widely used both in industrial and academic

institutions. PAFEC is available through the University network and can be accessed from any workstation or remote terminal.

Using PAFEC it is possible to perform investigations into thermal response (both transient and steady-state), static mechanical loading, dynamic response and crack analysis. Additional modules are available for lubrication, acoustic, electrochemical and fatigue analysis. The capability of the system is extensive and the flexibility within it is vast and consistent with almost twenty years of development.

At the time of commencement of this research, PAFEC was only available in its basic form with no pre- or post-processing facilities. In order to perform a simulation it was necessary to first form a text file, containing details of the model mesh (nodal co-ordinates and accompanying element topologies) and a list of 'control' commands, dictating the type of simulation to be performed and the restraints and conditions to be applied to the model. Output from the program consisted of text files containing the requested values (temperatures, stresses etc.) for the model and these had subsequently to be examined, analysed and any graphical representations devised by the user. Simulation runs could take between a few minutes and many hours, depending on model complexity, test type and system load.

The most time-consuming part of producing the input file is the discretization of the component into a mesh of individual elements. The qualities of the mesh are crucial to the success of the simulation and this is discussed in Section 3.3.3. The model input file containing mesh description and control commands can be hundreds of lines long. Sample model files are given in Appendix II.

Unfortunately, PAFEC contained no capability for the display of its results in simplified or graphical form and its output files containing results are massive for a model of any complexity. For example, for a model containing 3000 nodes in its mesh, no displaced shape plot was automatically available for representing distortion of a component, but a file containing individual nodal displacements in the three Cartesian directions for each node. This meant for any reasonable analysis of the results to be performed it was necessary to write programs, capable of extracting stresses, temperatures or distortions from the various results files and displaying them in a comprehensible format, usually with coloured contours representing the values superimposed on a three-dimensional solid model. Section 3.1.3 describes the

programs in more detail but this was a very important and time consuming part of the work, although in itself it had no use other than to display results.

The majority of simulations to be performed on brake components involved thermal distortion. In order for PAFEC to model this condition it is necessary to run the package twice. The first simulation is purely a thermal one, a transient analysis usually being necessary, to allow temperature distributions to be obtained for the whole of the model throughout the period of the test. The mesh contained in the model input file is defined using elements of a type designed for thermal analysis and any initial temperatures and *thermal* restraints (for example, any regions of the model that may be held at a fixed temperature through the test) are included in this. A suitable time step is chosen (see *Fourier's Equation (1)*, Section 3.3.3) and the program proceeds through the solution, calculating temperature distributions for each time step, until the specified maximum time is reached. The output from this run consists of temperatures at every node on the mesh for all the time steps defined for the transient analysis. This output is then utilised as the input for the second run. For this second run, the model file contains a geometrically identical mesh, but modelled with stress analysis elements, and any *mechanical* restraints that are to be placed on the model. The distorted form of the mesh is then generated from the calculated thermal strains (and any additional mechanical loads included in the model file), results showing displacements at each node. Various types of stresses can also be calculated for all or specified regions of the model.

3.1.3 Supplementary Programs

The requirement for the results of thermal runs to form the input for stress calculation runs and the lack of facilities for post-processing of results dictated that a number of additional computer programs were required to be written. In this way repeated use could be made of PAFEC without time-consuming and inaccurate manual retrieval of data either for further computation or graphical output. In the event many programs of varying complexity were written that created a system where simple input and output could be manipulated and adapted according to different simulations being performed with the least time and error. There were two main categories of programs.

The most important programs that were evolved dealt with the transformation of bare output text files and converted them to coloured representations of the model where

temperatures, stress and distortions could be assimilated at a glance. The output from these files appears much the same as other graphical output from other finite element packages with results post-processing capability. All the graphical results in this report were produced using the post-processing programs created for this work. In order to produce pictures of this quality and flexibility (any model can be viewed at any angle, contour scales can easily be changed etc.) UNIRAS subroutines were used. This library of subroutines contained ready pieces of FORTRAN code for a large number of applications from hidden line removal, shading and contouring to data manipulation (such as complex grid interpolation routines). The programs themselves are flexible. They utilise different PAFEC output files to produce the mesh definition and results and require little adaptation to take any model from PAFEC output and produce similar graphics. Although individual programs have to be called and run depending on the particular output required, (temperature, stress etc.) they collectively form a very acceptable post-processor for PAFEC.

The second type of programs that were developed for this work were concerned with data processing. Many of these were created to extract mesh information and results, and placing them in files of a format suitable for use by the post-processing programs described above. Others were required for generating models. For example, with some manipulation disc segment models (described later) could be converted into full disc models without the time-consuming manual data manipulation that would otherwise have been required. The majority of programs were straightforward to design and performed simple but long and tedious operations that enabled the production of different models with greater accuracy and speed than would otherwise have been possible.

Of all the programs none were dedicated to a particular model or type of model. Although each one had a single function all could be adapted by changing variables to perform that operation on different models and meshes of any size. A very efficient system was created, without which it would have been impossible to complete anywhere near as much practical experimentation. Appendix III contains a full list of programs written, their uses and filenames.

In addition to FORTRAN programs, spreadsheets were utilised primarily for mesh generation. In the instances where different models were required that were identical except for a varying dimension for example, it was relatively simple to construct a

sheet containing all node co-ordinates but where those that varied were linked to a single cell representing the dimension. When the contents of this cell were changed, the automatic recalculation facility of the spreadsheet recalculated the node co-ordinates required to change the particular dimension in the mesh. An illustration of this type of spreadsheet is given in Table 1. In this case it was used to generate models of varying rib width, the values being specified at the inner and outer edge of the braking face. This greatly improved efficiency in redesigning meshes for comparative testing as making alterations of this type manually is very time consuming. A list of these spreadsheets and their uses is also given in Appendix III.

	A	B	C	D	E	F	G	H	I	J	K
1	Node										
2	Number	X	Y	Z							
3	1	0.00	0.00	0.00	RIB WIDTH @ 88.50				RIB WIDTH @ 136.50		
4	2	5.50	31.75	0.00		5.00	mm		5.00	mm	
5	3	0.00	31.75	0.00							
6	4	8.50	34.75	0.00							
7	5	0.00	34.75	0.00							
8	6	8.50	56.40	0.00							

Table 1. Part of a model generation spreadsheet.

3.2 Strategy for Simulations

3.2.1 Necessary Simulation Capabilities

It was necessary to determine exactly what form the simulations were going to take, for example whether the brake disc would be modelled in two or three dimensions and exactly what conditions were going to be imposed on the disc. This was a vital stage in the determination of the simulation requirements and is detailed below.

It had initially been decided to make the study general in its approach, dealing with disc distortion behaviour only. The brief from Ford included a reasonably wide area of interest and bearing in mind there was only a year available in which to complete the work, it was necessary to take a broad view of the subject. To this end, although accuracy was necessarily vital, it was decided not to attempt to model the complex heat transfer effects at the pad/disc interface, but to use already proven knowledge to reproduce and extend common types of disc behaviour. Comparative testing was to be

employed in the interest of more straightforward modelling and quicker generation of useful results.

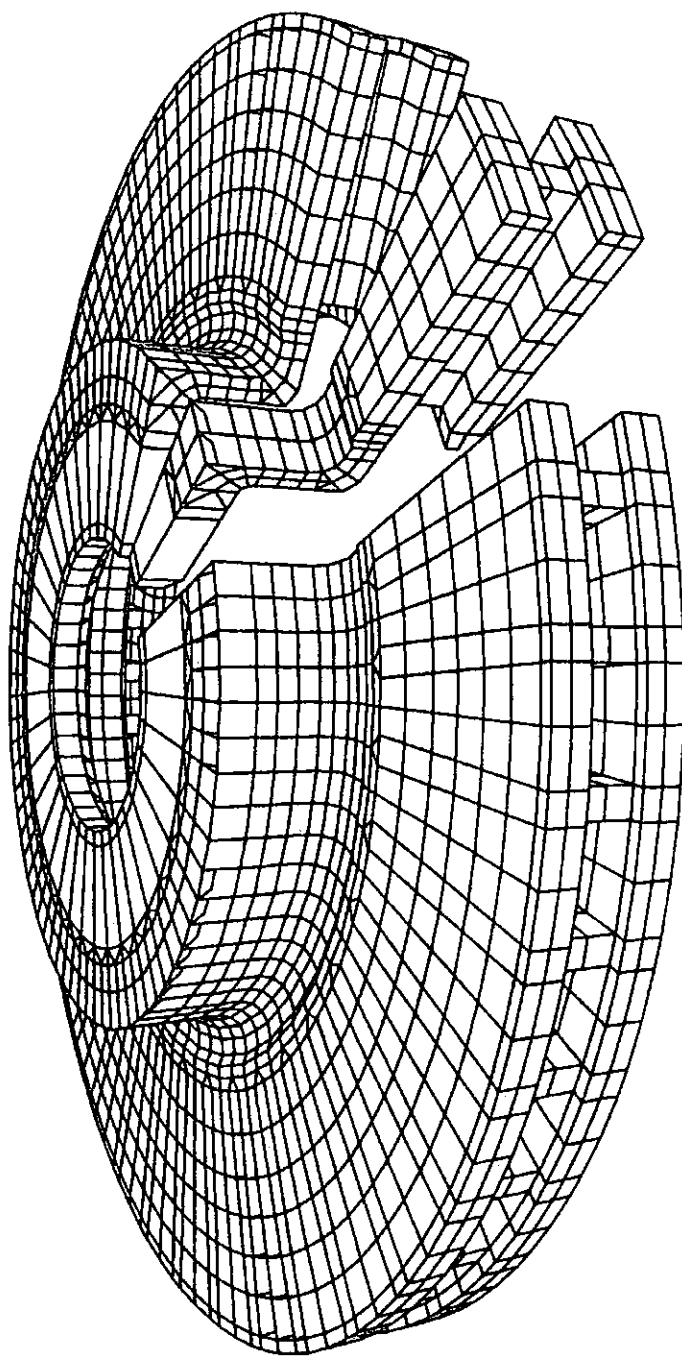
a) **Mesh considerations.**

The original Ford brief for the research called for the study of coning and waving, in addition to a comparison of conventional and 'inside-out' disc types. Waving, being a variation in the circumferential direction, necessitated the development of a three-dimensional full disc model but certain other disc characteristics, such as coning and radial expansion, have already been shown to be adequately simulated using segment models. The segment model, whilst still being three-dimensional, is only a model of a portion of the disc and takes advantage of the cyclic symmetry of ventilated brake discs. This is illustrated in Fig 7. along with a segment model. The use of segment models relies on the phenomenon being modelled, distortion, stress etc., to have no variation around the disc in a circumferential direction. Pure coning does not vary around the disc and therefore so long as the correct restraints are applied to the segment to constrain it to behave as part of a full disc then the segment model is valid. Waving is, by definition, a variation with angular position and so a segment model is inadequate and a full disc has to be produced. Segment models naturally have a much lower number of nodes/elements and therefore make a large saving in calculation time and cost. They are therefore desirable wherever valid.

If the disc is assumed to be perfect in form and mounted without runout then as far as heat transfer by conduction is concerned then not only are there lines of geometric symmetry but also lines of thermal symmetry. These are radial sections along which all heat flow takes place in a radial plane and across which there is no circumferential variation of temperature. The temperature distribution of the disc therefore also exhibits cyclic symmetry and providing there is equal heat input at any given instance to all radial positions on the braking face then a segment model is valid for thermal simulations¹.

¹ In the case of a solid disc with even heat input, an axisymmetric model can be used as the section of the disc and the direction of heat flow within it is identical at any given angular position. It is only the cyclic nature of the vanes of a ventilated disc that requires a segment model to be used instead of an axisymmetric model.

Disc with 37 ribs is comprised of 37 identical segments.



disc segment containing 1 rib.

Fig 7. Brake disc cyclic symmetry.

b) Model inputs

In order to successfully model brake disc behaviour it was necessary to define a test that would be applied to different discs for comparison. This test could also be applied to actual dynamometer discs and the results compared with simulation results. Ideally the test should be simple, brief and designed to produce severe disc distortion to aid accuracy in measurement and modelling. It could also be devised so that the effect of certain assumptions that would have to be made were minimised (discussed later). A single stop test was chosen. With suitable deceleration and simulated vehicle weight the test would produce severe results and would be short enough to ensure that disc cooling would not play a significant part in the disc response (see Section 3.3.1). The test would be performed with the disc initially at ambient temperature because of the naturally straightforward and known initial temperature distribution and because more severe distortion would be observed because of the greater temperature differences across the disc. Single stop testing also removed the necessity to study pad/disc wear during a test. The small amount of pad wear during a single stop would not affect the accuracy of a simulation of general disc behaviour. It was hoped that the test would yield consistent and repeatable results because of its simplicity and severity. Details of the test conditions, arrived at after discussion with Ford brake design engineers, is given in Appendix IV. This test would form the basis of computer simulations performed in Durham in addition to dynamometer testing carried out at Ford for comparison of results.

Other tests were also derived such as simple drag tests and cooling tests but these were arrived at after an amount of simulation work had been completed and are described in greater detail later. For the most part they were simple and designed to demonstrate certain effects as opposed to producing detailed and accurate data on disc behaviour.

It had originally been intended to use the results of thermal imaging tests, carried out at Ford, as input into the disc models in much the same way as Inoue [18]. This is an ideal way of utilising finite element models as the use of measured temperature maps removes the requirement for assumptions to be made about the exact nature of heat input into the disc which usually form the greatest source of error in a simulation. The model distortion results are much more likely to bear a close resemblance to the results obtained from the dynamometer or vehicle disc used to provide the input than if the heat input is approximated. However the installation and validation of the equipment

took much longer than had been estimated and there was not sufficient time to wait until it was completed. It was therefore impossible to use thermal imaging data as part of the input to the simulations.

Because the brief of the project had been the study of disc behaviour in a fairly general sense it was not thought necessary to model marginal effects and try and produce absolute results predicting small quantities. Therefore it was decided not to produce detailed predictions about heat transfer and macroscopic material changes at the friction interface. Similarly wear is very complex and required more time than was available to model satisfactorily. Because such effects were unnecessary and not within the scope of the brief it was decided not to include them in this work and the single stop test was devised with the aim of reducing the effect that they would have on a simulation. For this same reason it was decided not to utilise an iterative solution type. This would have taken too long and although undoubtedly more accurate, as shown by Day et al. [13], it would not be necessary (and therefore a waste of valuable time) to incorporate it in research into general disc behaviour as opposed to detail friction interface work. It is important to stress that general disc behaviour is being studied (more work can be accomplished in a time span if it is kept simple) and all testing is comparative. In this way certain inaccuracies need not be worried about because they are effectively cancelled out, or at least their effect is greatly reduced, when one similar test is compared with another, the difference in the two results being used to form conclusions about component behaviour. Even small inaccuracies in values used in the simulation have a reduced effect on the validity and confidence of the results. This is important in understanding the approach taken to this work, allowing a large number of different tests to be accomplished with confidence because of the careful selection of the subject matter of the modelling and the tests used in the simulations. More on this subject is included in Sections 3.3.1 and 3.3.2.

This having been said, the pad pressure distribution does play a governing part in the heat input into the braking faces of the disc and, as has been previously stated, it was felt that the traditional assumptions of heat input being a) constant over the radial dimension of the braking face and b) uniform around the disc face (and occasionally constant with respect to time!) were wholly inaccurate and therefore unacceptable. In this case, because of the caliper being a single piston type, the load distributions on the backplate of the pad were very different from one side of the disc to the other. This would doubtless have an effect on disc response also.

This meant that some study of pad behaviour would also be required and therefore it was planned to produce an additional finite element model of the brake pads, subjecting it to different loads according to which side of the disc it was mounted on. In addition to the model a program would be required to convert the friction pad face pressure distribution obtained from a pad test (using the single stop test conditions) into suitable thermal input data (and normal and friction loading on the disc face) which could then be used to achieve a disc response for the single stop test. In this way it was hoped that a far more accurate description of the thermal input to the disc would be achieved with some investigation into such effects as heat rings and spots that result from uneven pressure/heat distributions. Although pad pressure distributions have been investigated by several authors no papers were encountered in the literature search which made a specific link between the distribution obtained from modelling a specific pad and applying to the corresponding disc, using it to perform a study of the resulting behaviour of the disc.

As it had been shown by Tirovic and Day [25] that caliper flexure did not produce a significant effect compared to that of varying pad pressure distribution, it was decided not spend the considerable amount of time that would have been required to produce a model of the brake caliper and incorporate this aspect into the simulations. There would be no significant loss of accuracy, particularly in the face of other, much more significant approximations.

3.2.2 Summary of Models for Development

After a period of investigation it was decided that the following outline would be implemented.

A pad model would be constructed and loads appropriate to the chosen single stop test and its position in the caliper applied to it. The resulting pad pressure distributions would then be extracted and used to calculate a temperature map for the disc surface that varied with time appropriate again to the conditions of the stop test (vehicle speed and therefore pad/disc speed etc.). This would form the basis of input to the disc models and once obtained could be used each time as an accurate and consistent input to the simulations. Mechanical loads, both reaction and friction, could also be obtained using the pad test results and applied to the discs in the second, PAFEC (disc displacement and stress calculation) run.

Two principal model types would be developed, a full disc and a disc segment, according to the specific simulation to be run. Thermal inputs obtained above would then be applied to the equivalent thermal model mesh, a thermal PAFEC computation run made and the results, combined with the mechanical loading mentioned above and physical restraints, used in the second displacement/stress PAFEC computation run with the equivalent stress model mesh. The results of this simulation would then be extracted using data processing programs written for this purpose and graphical output produced using post-processing programs, again written expressly for this purpose. Different changes would then be made to the geometry of the disc and the single stop simulated again and used in comparison with others of similar input conditions. A definitive simulation run would be performed of the current disc design to act as the reference for future tests.

3.3 Model Generation

3.3.1 Assumptions for Model

The approach taken in this work, that of studying general disc behaviour, has already been established and this approach is defined through the assumptions made in the establishing of the conditions for the finite element simulations. The major considerations are outlined below.

1. 'Segment' vs. 'Disc' - Inoue [18] wrote that, providing a heat input is applied that is identical at all angular positions on the braking face then no distortion or thermal phenomenon occurs that varies around the disc. Coning and radial expansion could be examined this way, even though it may not be strictly correct in practice, because the existence of these phenomena is not dependent on or affected by circumferential variation. As much of the work concerns these effects then a considerable time saving achieved by using a disc segment model instead of a full model would be desirable (a segment model is made to represent a full disc by the application of special restraints at the boundary and the validity of this is tested by comparison of the results obtained by a segment and full disc subjected to the same even heat input. This is described in Section 4.2.3). A full disc model with circumferentially varying heat input can be run near the end of testing to confirm the observations made using a segment disc.

2. Importance of cooling - Although convection cooling has an important effect on disc behaviour it has been shown that during a single stop, cooling by convection has a minimal effect on disc temperatures. This was extensively investigated by Newcomb [2], who concluded that '*[disc] surface temperatures reached during a single [brake] application are little affected by air convection losses from the disc surface*'. This is because of the initial low temperature of the disc, dictating that far more heat can be absorbed into the body of the disc by conduction owing to high temperature difference between ambient and those instantaneously achieved on the braking face at the friction interface, than can be lost by convection. This assumption has been demonstrated previously to be valid and is examined again in Section 4.2.4.
3. Constant hub temperature - The disc is mounted to a large mass of metal initially at ambient temperature and so, assuming good conductive characteristics at the interface (large metal faces under significant contact pressure form this interface), the assumption that the hub of the disc where it contacts the axle remains at the same temperature throughout the test is valid. For simplicity of modelling it is assumed that the interface between the hub and its mounting to the vehicle axle has perfect thermal conductive properties. It can therefore be thought to act as a heat sink, never deviating from ambient temperature. The single stop test was chosen again because of its short duration, during which most heat is absorbed into the braking face and there is negligible heat flow out of the disc at the hub interface anyway.
4. Constant heat input over each time step - It was assumed that because of the disc surface speed being high (especially in the initial stages of braking where most of the heat is generated) heat input could be considered even over small intervals of time (0.25s was eventually chosen as the time step size - see Section 3.3.3). Therefore, for each of the time steps in the transient analysis, the heat input is assumed constant. Similarly, during these time steps the heat input around the face of the disc was considered constant.
5. Segment pad loads - Tirovic & Day [25] wrote '*[their work] suggests that disc deflection due to mechanical loading alone is small compared to thermal loading*'. Pad loads were not included in the segment model for this reason. In addition pad loads do not exhibit cyclic symmetry and so would be inappropriate

and inaccurate in the segment model. It was, of course, necessary to include them in the full disc model, both for accuracy but also because the pad/disc mechanical behaviour is thought to contribute to the effects of runout and waving.

6. Constant material properties - Virtually all material properties change with temperature variations. Incorporation of this fact into a finite element solution would require an iterative solution which greatly increases the time taken for the computations. The anisotropic nature of friction material properties has been well documented [27] but because thermal effects are not being considered in the pad, the inclusion of them is not necessary. The effect of material changes on the disc is also sufficiently small not to be a significant source of error.
7. Friction film - As has already been extensively discussed the detailed heat transfer conditions are not being modelled and constant frictional properties are being assumed, although in practice friction coefficients and behaviour varies with both temperature and pressure variations (caused by runout, previous deformation etc.). In the case of this particular friction material values of between 0.35 and 0.55 are possible, depending on the age of the material and the temperature/pressure conditions. The assumption of mean values of coefficient of friction for the tests, derived after consultation with Ford brake design and test engineers, does not present significant inaccuracy. In practice, perfect pad/disc contact is assumed, analogous to a fully bedded disc/pad, and calculations based on a constant friction coefficient over the stop. Fade was not to be specifically considered.
8. Constant deceleration - For this simulation a constant deceleration is assumed for two principle reasons. Firstly, it is far more simple to produce a computer model based on constant deceleration and secondly it is more simple to do dynamometer tests using a constant resisting torque to simulate braking of a moving vehicle. The assumption of constant deceleration is not strictly accurate, variations during an actual stop coming principally from variations in pad μ (brake 'fade' being an obvious example) through the stop. The assumption of a constant pad μ (see above) necessitates the assumption of a constant braking torque and therefore a constant deceleration. The friction film mentioned above might also affect the assumption that heat input is proportional to pad surface speed, and therefore vehicle speed, and decreases linearly with time during a constant deceleration.

constant deceleration. The friction film mentioned above might also affect the assumption that heat input is proportional to pad surface speed, and therefore vehicle speed, and decreases linearly with time during a constant deceleration.

This, again, does not present great inaccuracy because of the nature of the stop tests being simulated.

9. Centrifugal forces - A simple and accurate assumption is that the stresses induced in the brake disc because of its spinning motion are negligible. This is demonstrated in the model validation and described in Section 4.2.5.
10. Rigid disc - For the purposes of the pad face stress distribution model it was assumed that the disc remained rigid. This had previously been investigated by Tirovic et al. [25].
11. Pad backplate/friction material interface - Although there is no evidence to the contrary it is necessary to assume the bond between the pad backplate and the friction material block is rigid. The quality of the modern, purpose-designed adhesives employed ensure that this is a negligible error.

3.3.2 Limitations of Model Accuracy

It was accepted that there would be limitations to the accuracy of the model caused by time available and allowed model complexity. Traditionally the greatest of these concerns the definition of the heat generation inputs. Although steps have been taken to ensure that both the stop test and the level of modelling has been defined to an accuracy consistent with the type of desired results, errors will occur.

It has not been possible to account for the effects of disc distortion on the pad pressure distribution and therefore heat input during the stop. The simulation is to be conducted with the same radial heat input distribution over the braking face throughout and although the effects of coning on this can be deduced from the results, this can not be included in the simulation itself and quantified. In addition no measure of the minor effects of caliper flexure and sliding friction are available for this specific case.

Mesh quality (mesh construction described in the following Section) also has a significant effect on the error of the numerical results and although there are established techniques in mesh design and ways of checking the quality of a mesh with respect to its solution, error does exist. Steps were taken in the model validation phase to ensure that the most efficient combination of numbers of elements and numbers of nodes was employed and that the resulting mesh was suitable. However an amount of error, very difficult to quantify, will have arisen.

3.3.3 Mesh Generation

The finite element mesh is the heart of the Finite Element Method. It is therefore very important that it is well designed. The number and type of elements (and therefore the number of nodes) and the grading of the mesh (the variation of the element size throughout) are all crucial and guidelines exist for the determination of each. Quadratic elements contain a midside node on all element edges, whereas linear elements only possess corner nodes, and their use results in greater modelling accuracy (more 'Degrees of Freedom' per element). On the whole the greater the number of nodes and elements, the more accurate the results the mesh will yield. This is analogous to approximating the area under a curve with a number of trapeziums, the greater the number of trapeziums, the more accurate the estimation of the area. However the effect also has a diminishing return, such that less and less advantage is gained by increasing nodes (until error actually increases because of the rise of a particular calculation error). An increase in number of D.o.F.'s results in a corresponding increase in computation time and therefore to maintain an efficient solution system the number of D.o.F.'s has to be carefully chosen. Other errors can be incurred due to the excessive distortion of the prismatic elements (usually brick or triangular) as they are employed in the mesh and PAFEC produces warnings of distorted elements when analysing the quality and integrity of the mesh submitted for analysis. In general it is advisable to use larger elements where stress concentrations are low and smaller elements in a denser mesh where stress concentrations are high for combination of solution efficiency and accuracy. The effectiveness of the mesh in this respect is quantifiable within PAFEC and this is included in the model validation. Whilst defining this variation in mesh size it is also advisable to maintain as smooth as possible the rate of change of size from one element to the next.

Disc: Several attempts were made at producing disc meshes and two of these are illustrated in Fig 8. The meshes required for temperature and stress models are required to be identical and were generated by taking form and dimensions from the engineering drawings for the disc and subdividing into elements, maintaining accuracy of geometry and incorporating the rules outlined above. Experience plays a large part in the design of an accurate mesh and the results of the previous study were used in determining where the finer areas of the mesh should be. Larger elements are employed where stress and temperature variations are not so great, the same mesh topology needs to be employed for both temperature and stress calculation runs. Many basic prismatic elements were available for fitting the disc section, with the same geometry but different calculation properties for temperature and stress analyses. The final meshes chosen to form the disc segment models (both conventional and inside-out) and to form the basis of the full disc models are shown in Fig 9. They each contain approximately 1200 nodes and 175 elements, a total of 3700 Degrees of Freedom. Trials were conducted during model validation using models of identical topology but with combinations of varying node and element numbers (see Section 4.2.1). The results were compared with each other and the above combination of nodes and elements was chosen as giving results practically identical to those obtained by much finer meshes with many more nodes, yet possessing a much faster solution time. In generating this mesh it was important to observe Fourier's Equation (1) in the sizing of certain elements. PAFEC uses the Crank-Nicholson method of obtaining a transient solution by way of iterative, linear time steps. By the nature of the method the results at the end of each time step tend to oscillate about the true solution, the amplitude of this oscillation decaying or increasing with time depending on whether the solution is stable or unstable. It has been found that the stability of the method is governed by the variables within the calculation, and in particular that the method will be stable with a minimised amplitude of oscillation if the variables are chosen such that the dimensionless Fourier's Number, F , is equal to one. In practice, this involves calculating a suitable time step according to the size of the elements undergoing the most severe thermal shock (in this case, those on the disc braking faces) and means that the solution will tend to converge and that the results will be more accurate in the early stages of the analysis. The mesh was refined over a period to obtain the most accurate version, no warnings of distorted elements being given by the software.

Fourier's Equation

$$F = \frac{k \cdot \delta t}{Cp \cdot \rho \cdot \epsilon^2} = 1 \text{ if stability is to be maintained} \quad (1)$$

Using this equation a transient time step value of 0.25s was chosen as the most desirable value. This value was low enough to maintain good definition of the variables within the simulation, whilst giving an excellent mesh grading at the braking face on the model. The resulting Fourier's Number was 0.98. As the stop time for the test is 7.5s, 30 transient steps were required for each simulation run and this quantity is more than enough to ensure the validity of assumption #4 (Section 3.3.1) concerning even heat input over time steps. In addition, with a well chosen Fourier Number the amplitude of oscillation, and therefore the calculation error, will be small after this number of time steps. For maximum accuracy, results are usually compared at the end of the transient calculations.

Pad: The mesh designed to model the pads is shown in Fig 10. It was designed using the same criteria for quality as the disc section mesh. Fourier's equation was not necessary as thermal loads were not being applied to the model.

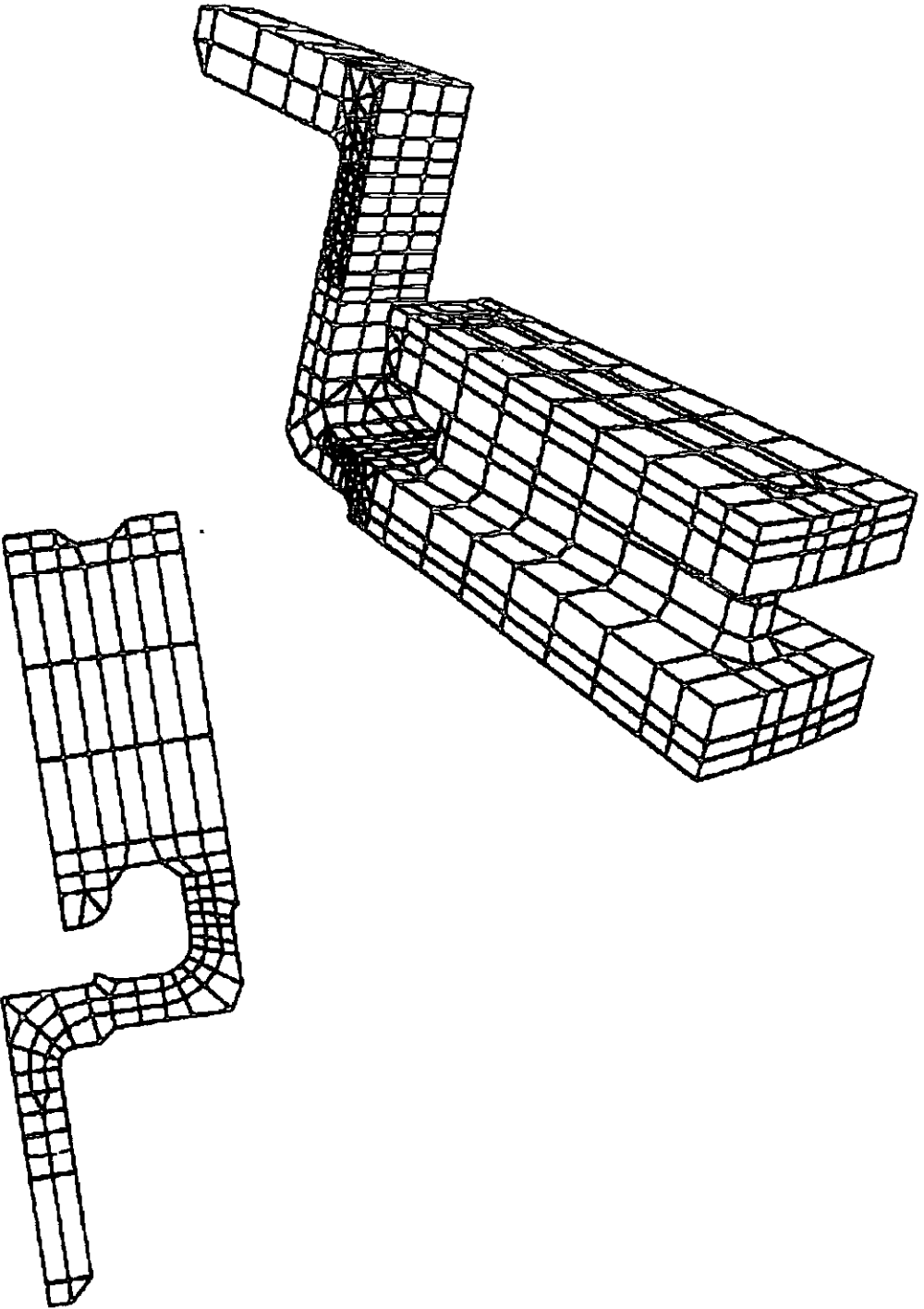


Fig 8. Preliminary disc segment model meshes.

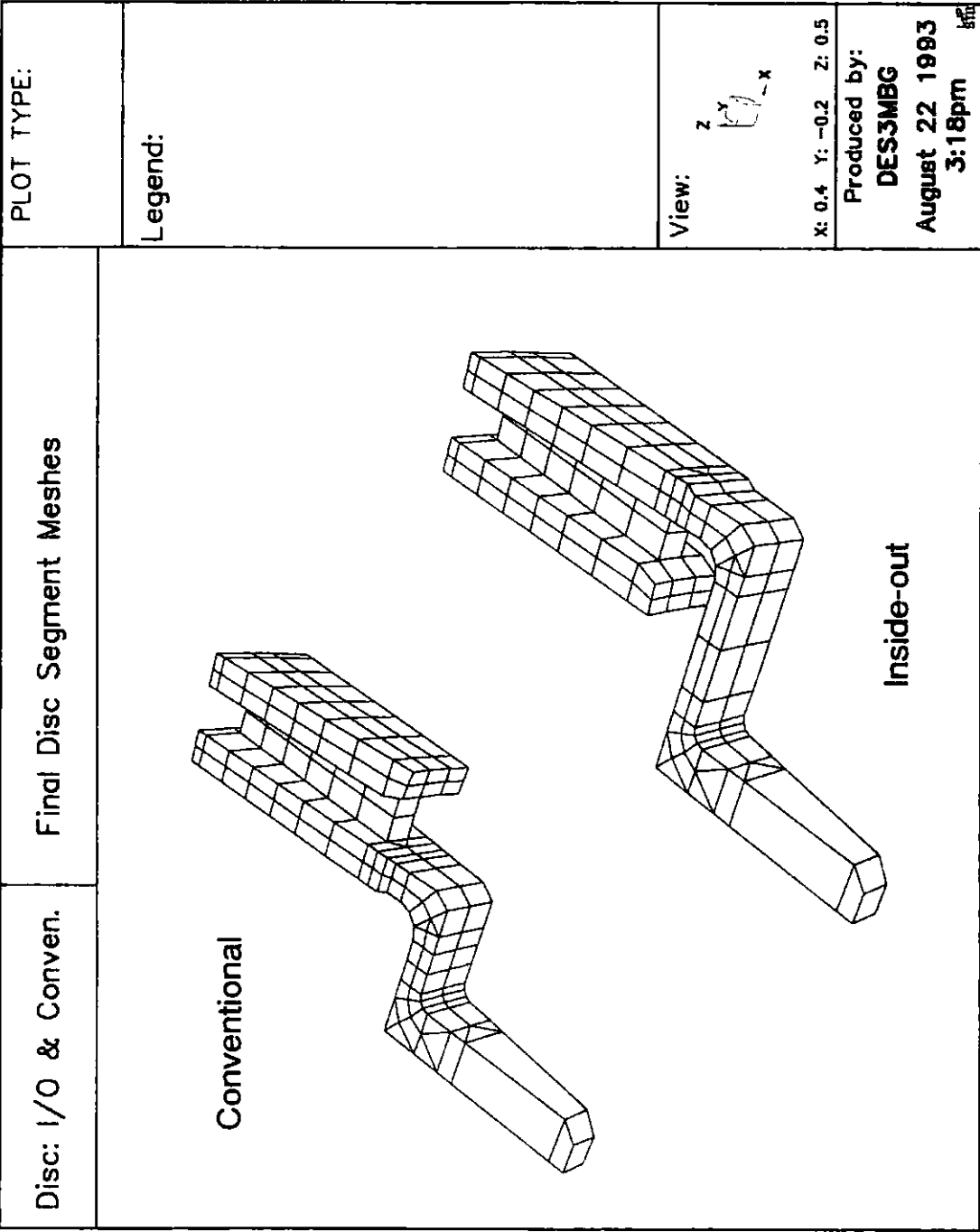


Fig 9. Chosen disc segment model meshes.

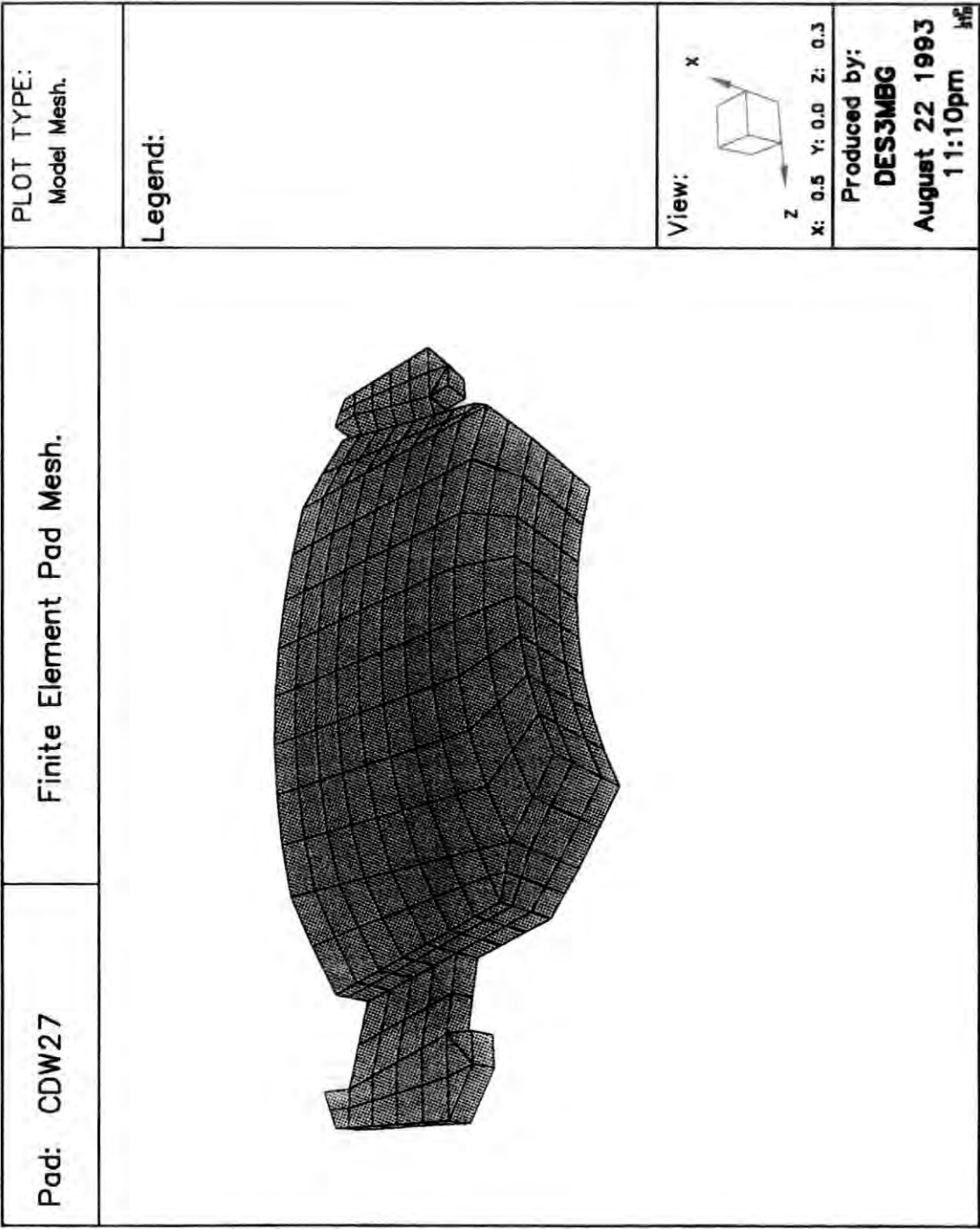


Fig 10. Pad model mesh.

3.3.4 Model Restraints

In order for the mesh to accurately simulate the behaviour of the component it has been designed to model it is necessary to impose both thermal and physical restraints. These are required to fully describe the fixed conditions of the model, in other words the quantities that remain constant through the simulation. Holding displacement to zero at a certain part of the model is analogous to a rigid connection to ground over this region and similarly maintaining a constant temperature equates to that region of the object in good thermal contact with a heat sink of that temperature. According to the specific details of the restraint different constraining effects can be applied such as rotation without translational displacement. It is therefore important that these restraints are applied carefully and fully or unexpected behaviour is likely to occur.

- a) Pad: The loads being simulated in the case of the pad were purely mechanical and consisted of a normal actuation load on the backplate of the pad representing caliper cylinder/reaction, and a frictional load on the face of the friction pad itself. The reaction to these forces were supplied by the disc face and the caliper body bearing against the lugs on the side of the pad respectively. After much consideration involving the techniques that could be employed within PAFEC to model this, it was decided to fully restrain the face of the pad (a rigid disc being assumed - [25]) and apply fixed displacements to the backplate to simulate the depression caused by the cylinder/caliper fingers and laterally to the lugs on the side of the pad. PAFEC was able to calculate the forces produced by the fixed displacements in the structure and these displacements were chosen such that the reaction and friction forces that resulted equalled those known from the data supplied by the brake manufacturer for the stop conditions being modelled. The reaction on each lug was a proportion of tensile or compressive force, dependent on the lug and the pad being modelled. The pattern of displacement in the back of the pad was again dependent on the pad being modelled, the caliper fingers and the cylinder having different contact areas with the backplate. Fig 11. illustrates the location of the fixed displacements.

Pad/Caliper Contact Areas

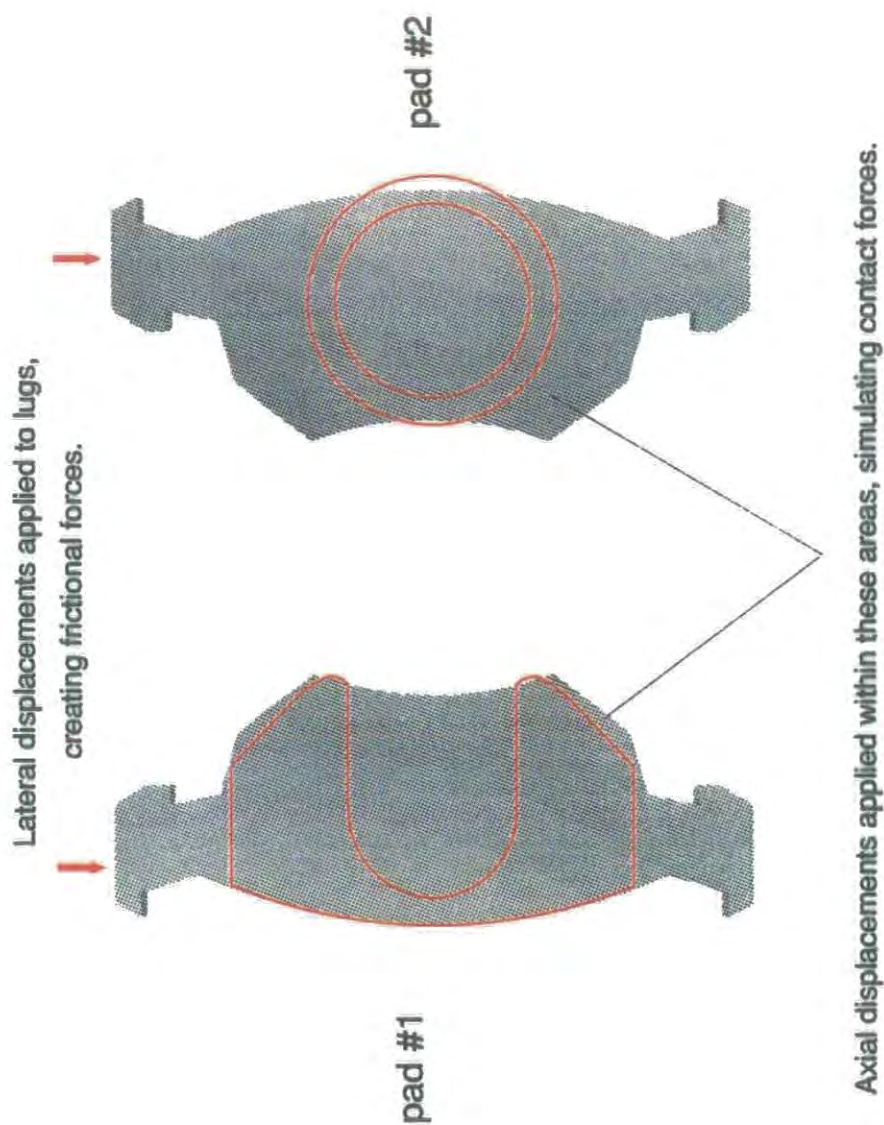


Fig 11. Caliper/pad backplate contact areas.

- b) Disc - thermal: In the case of the thermal model of the disc, the temperature difference across the disc is caused by the hub remaining at a constant temperature whilst the braking face is rapidly heated by friction. This temperature is held constant (ambient in this case) by the fact that there is a large mass of metal (axle, wheel etc.) in good thermal contact with the disc hub. To simulate this the region of the hub in contact with the axle was restrained to ambient temperature. This was the same for both the segment and full disc models. No thermal restraints were required to simulate full disc behaviour in the segment because by careful selection of the segment size and position all heat transfer by conduction was known to be radial on the boundary sections on either side of the segment mesh.
- c) Disc - stress: The only mechanical restraint on the brake disc is its rigid mounting, again at the hub. In service the disc hub is tightly 'sandwiched' between two annular faces and this was simply modelled by fully constraining all nodes on the faces in contact with the mounting faces. This was the case for both full disc and segment models. However it was necessary to impose further restraint on the segment in order to simulate behaviour as part of a full disc. Because of the inherent symmetrical nature of the segment it is apparent that the sections on the either side of the segment (faces of constant angular position) would not move from that plane in the event of uniform disc expansion. Therefore if these faces are constrained, only allowing free movement in the axial and radial directions the segment is behaving in the same way as a segment of a true full disc. Of course, when waving and brake pad loads are incorporated this no longer holds true and a full disc model is required but whilst simple expansion and coning is being examined and the effects of pad loads are not included (Day et al. [25] found them to be small) it is absolutely valid. Fig 12. illustrates the mechanical restraints applied to the segment models and their respective directions.

Segment Displacement Restraints

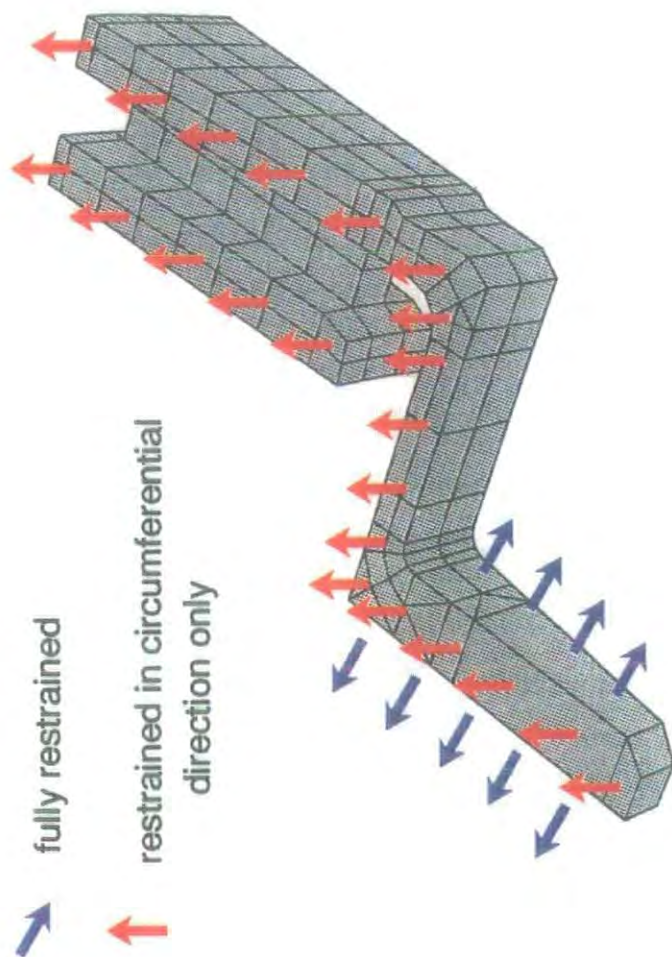


Fig 12. Model restraints.

3.4 Thermal Input Generation

The first step in the calculation of the thermal inputs was to run the brake pad model twice, once for each of the two pads in the caliper using the specific load contact areas in each case. Using a program developed especially for this purpose ('stressprocseg1.f' - see Appendix III) the pad face stress distribution was extracted from the results file and held in an array relating the co-ordinates of the pad face node and the axial stress (local pad face pressure) at that node. This stress map was interpolated into a fine grid of values (using a UNIRAS subroutine). This grid was then divided into eight radial bands corresponding to element bands on the braking face of the model mesh. In this way an accurate profile of pressure across the braking face in the radial direction could be obtained. Calculating the velocity of the disc relative to the pad at the centre of each of the radial bands and using this velocity, the coefficient of friction and the pad pressure allowed an estimation of the heat input to each band over a revolution of the disc. The following equation shows the relationship used.

Friction Heat Generation

$$\dot{Q} = \mu \cdot P \cdot v \tag{2}$$

The velocity for each of the thirty 0.25s time steps was calculated and it was then straightforward to calculate the average heat input to each radial band for all the 0.25s time steps and using a complicated value weighting method these values could be mapped onto the nodes forming the brake face in either segment or full disc model meshes. It was necessary to account for the partitioning in heat energy between the friction pad and the disc and the following equation (3) derived by Newcomb [1] was employed to calculate this ratio. γ is the ratio of heat energy entering the disc to the energy entering the pad and is essential in the calculation of the disc temperature rise caused by heat energy entering the disc. The value of γ was found to be low (about 0.05) and therefore unlikely to be a large source of error in itself.

Thermal Partitioning Equation

$$\gamma = 1 / \left[1 + \left\{ \frac{\rho_p \cdot C p_p \cdot k_p}{\rho_r \cdot C p_r \cdot k_r} \right\}^{\frac{1}{2}} \cdot \frac{A_p}{A_r} \right] \tag{3}$$

The results of this program were checked repeatedly and a high level of confidence achieved in its ability. It was written to be very flexible and allowed differing velocities, friction coefficients, material properties and many other test details. This program is at the heart of the technique of using pad face stress distributions to form accurate descriptions of heat energy input to the disc and this process would have been impossible to accomplish manually.

3.5 Simulation Runs

3.5.1 Overview

The first model runs to be made were concerned with the validation of the segment and full disc models. These runs involved comparison of results between the two models for identical inputs in attempt to gauge the ability of the segment model to reproduce the results of the full disc. In addition other runs were made to assess the contribution of cooling and centrifugal stresses for example, to ensure that the assumptions made earlier were valid. Once the models had been confirmed as accurate they were used to provide a datum record of the predicted behaviour of the existing design of disc ('inside-out'). Following this a conventional version of the same disc, identical dimensions except for the section where the hub meets the braking face, was developed and a similar procedure carried out as a reference for conventional disc designs. It was already presumed that much work would be performed on both types of disc for comparison. This phase of experimental work is documented in Section 4.

The results of these two cases were used to form hypotheses about the mechanism of some forms of distortion and using these hypotheses variations of disc design were modelled and tested against each other and the original results. Following a logical progression of investigation and then design revision based on results, a series of simulation runs was undertaken and the results of these are recorded in Section 5.

3.5.2 Creation of Disc Models

Variations in disc design and therefore model mesh were achieved with the use of spreadsheets as previously outlined. The spreadsheet would be altered, the relevant

portion copied into the PAFEC input text file and the model rerun, results being extracted in the usual way. The same thermal inputs, calculated from the pad stress distribution, were used in each case as the standard, consistent input. In most cases, five different models based on the variation of a single dimension would be created and all simulations run in a batch. Appendix II shows examples of the model input files.

3.5.3 Details of Simulation Runs

It was decided that the simulation runs would be a logical progression of tests, designed on the basis of the observations gained from simulating the current 'inside-out' disc and the conventional design. In addition each test could incorporate observations resulting from the test before it and to some extent an 'evolution' of ideas would develop, following different paths. The simulation runs are therefore discussed after the validation of the model, in Section 5.

CHAPTER 4

MODEL VALIDATION

4.1 Pad Test

The two pad models were run and the results for the two cases are shown in Fig 13. Pad #1 acts on the outer disc face as it is mounted in the vehicle, pad #2 on the inner.

It is clear that substantially different and uneven heat input patterns will result, there being in the case of pad #1 a ratio of 3:1 of the maximum to minimum face pressure across the pad. Pad #1 has a marked band of high pressure at the outside edge of the disc and the effect of this will be compounded by the higher disc surface velocity at this point. Pressure distribution is crucial in determining the thermal inputs and even though the same total force (average pressure) may be present in each case, the variation of disc velocity over the braking face of the disc can exaggerate the effect of any non-uniformity. The results of these pad models, clearly show that it was necessary to create these models, a uniform approximation being insufficient.

These results were then processed using a FORTRAN program and heat fluxes for the disc faces were generated and used for all subsequent test cases.

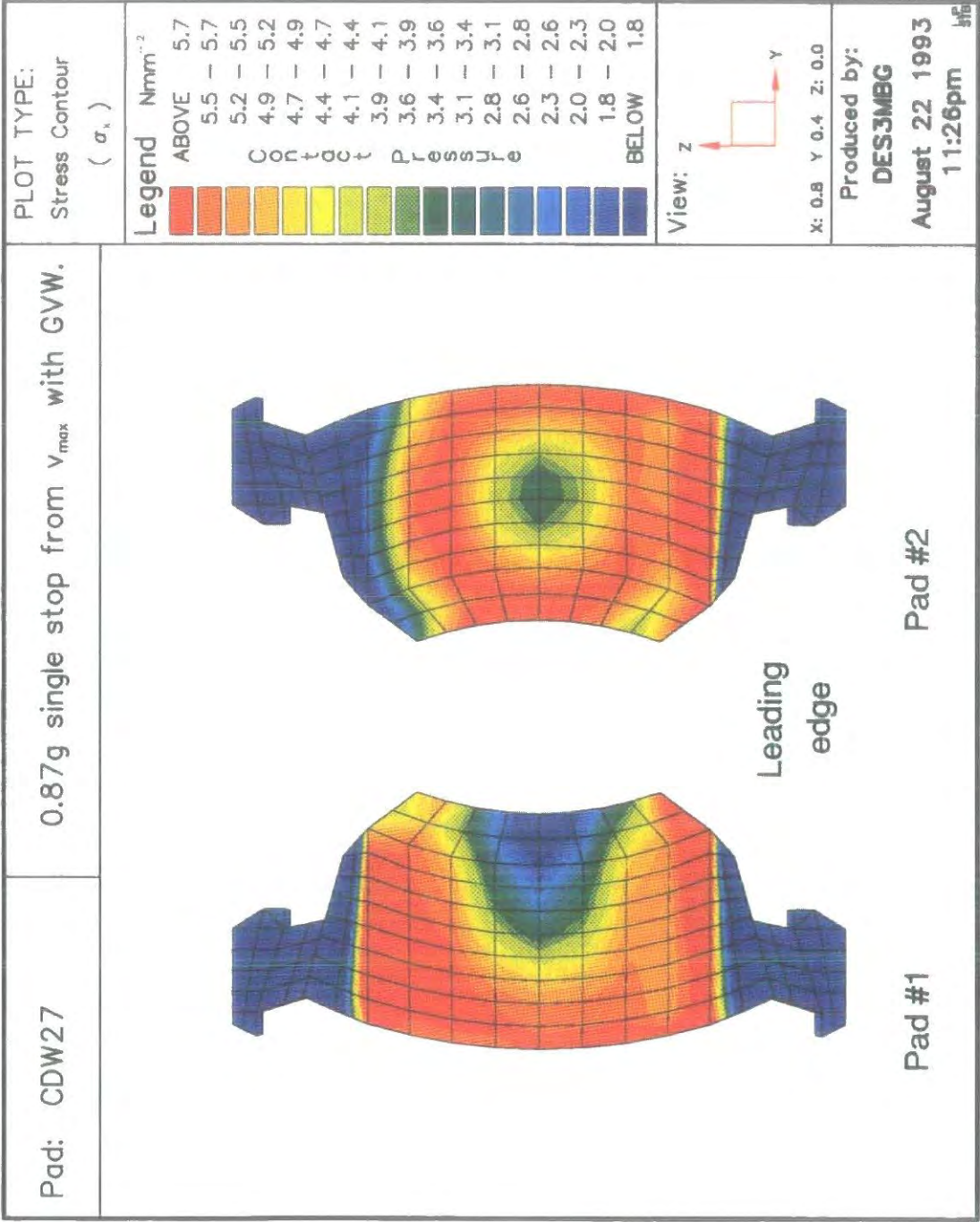


Fig 13. Pad pressure distributions.

4.2 Validation of the Disc Model

4.2.1 Degrees of Freedom

The first simulation consisted of developing disc models of identical geometry but differing numbers of nodes and elements. The tables below show the four segment models and the two full disc models created.

Model	Number of Nodes	Number of Elements	Degrees of Freedom	Element Type
A	362	177	1086	linear
B	1231	177	3693	quadratic
C	2007	1380	6021	linear
D	7347	1380	22041	quadratic

Table 2. Segment models.

Model	Number of Nodes	Number of Elements	Degrees of Freedom	Element Type
A	9759	6549	29277	linear
B	35769	6549	107307	quadratic

Table 3. Full disc models.

The models were re-run and the graph in Fig 14 shows a sample result varying with the number of D.o.F.'s contained within the model. For the definitive segment model, 'B' was chosen because there was no appreciable loss of accuracy and calculation time was economical. For the full disc, it was necessary to use linear elements ('A') even though more error was incurred because of the prohibitively long calculation time required for so many D.o.F.'s in a quadratic model.

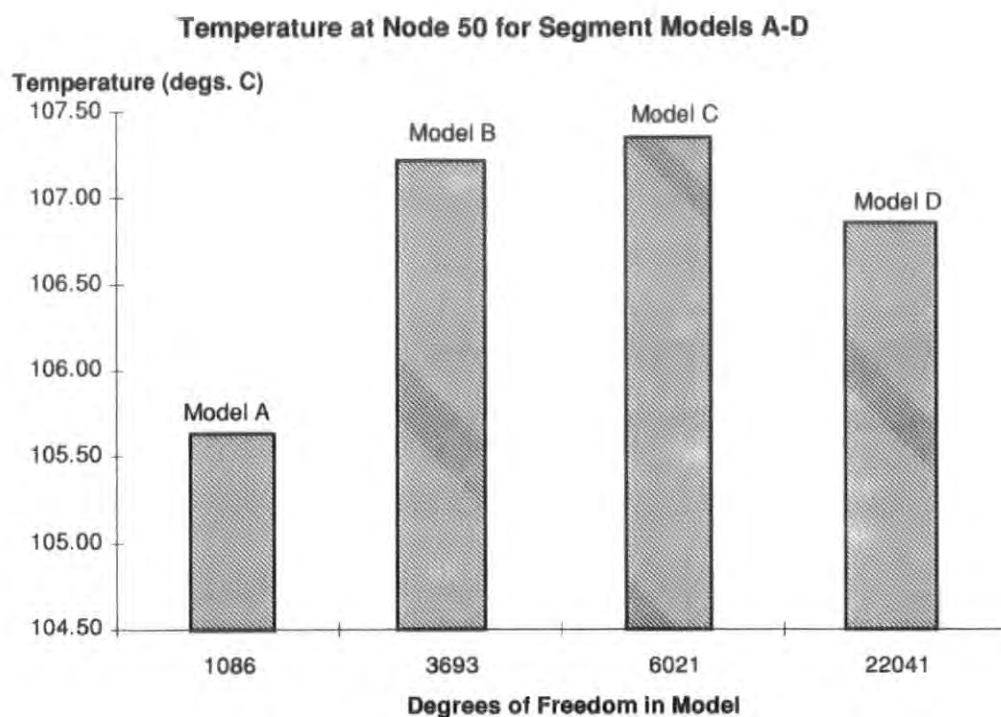


Fig 14. Variation of output value with model D.o.F.'s

4.2.2 Pad Loads

This simulation was run to test the assumption that mechanical loads imposed on the disc by the pad during braking were small enough to be discounted. Three models were run.

1. Thermal loads only.
2. Mechanical pad loads and thermal loads.
3. Mechanical pad loads only.

Two nodes were selected, one with very high stress values and the other with high axial displacement, and these values compared across the models. The results are shown in the table below.

Run	Von Mises Stress (N/mm ²)	X Displacement (mm)
1. Thermal loads only	650	1.035
3. Pad loads only	40	0.110

Table 4. Comparison of mechanical and thermal loads.

In addition, all displacements at an arbitrarily selected node were compared.

Run	X Displacement (mm)	Y Displacement (mm)
1. Thermal loads only	0.318	0.855
2. Thermal & pad loads	0.417	0.828
3. Pad loads only	0.098	-0.026

Table 5. Node 62 displacements.

If these results are examined closely it can be seen that there is a superposition effect. All displacements on the model were found to conform very precisely to this superposition. From this it can be said that if the segment is tested without pad loads the same thermal coning will result and can be studied independently. As in all cases of displacement the value caused by pad loads alone is significantly less than that caused by thermal loads it is appropriate to model the segment without using pad loads. The stress, measured at the stress maximum position on the disc is also considerably less with pad loads alone than with thermal. However, they are not negligible and it must be borne in mind, at least for the time being, that thermal coning and expansion is all that is being studied and not complete disc response.

Pad loads alone tended to cause slight concavity in the disc faces because of the central bias of the pad pressures used in the simulation (the actual pad model results were not used in this test - this particular run was carried out before the pad model had been fully developed) and the stresses caused by this result in very slight coning of the disc.

A Strain Energy Density module was included in this model run to provide a measure of mesh quality. S.E.D. is used in PAFEC as an indication of where a mesh might need refining for a specific solution to be performed accurately. Values of the same order

were returned for practically all the elements in the mesh indicating that the mesh grading was suitable to cope with the variation of stress throughout this particular model.

4.2.3 Segment/Full Disc Correlation

Two simulation runs were also performed with the aim of comparing the results obtained by a full disc model and a segment model for the same heat inputs. The same conditions were applied to both models and both thermal and mechanical simulation runs made. The greatest discrepancy between temperatures at any similar point on the two models was 1% and the equivalent discrepancy for stresses was 7%. The modes of deformation and the distributions of temperature and stress were identical for the two models. The error obtained was more likely to be the result of using a quadratic segment model and a linear disc model, necessary as already explained, than any inability of a segment model to accurately represent a full disc.

4.2.4 Face Cooling

Model runs were performed to test the assumption that it would be unnecessary to include convection cooling effects in the models because of the duration of the single stop test. Cooling equations for the braking faces of the disc were obtained from Sheridan et al. [19] (their derivation is given in Appendix V). The cooling effect of air within the ventilated portion of the disc was not included because of the lack of empirical data for 'inside-out' discs, but this is likely to be far inferior to the cooling of the external brake faces because of surface temperatures and cooling air velocity.

The maximum discrepancy between temperatures at the same points on the models (one cooled with convection, and the other not) was 0.5%. It was therefore decided that it was not worth incorporating it in the segment model because of the additional computation time.

4.2.5 Centrifugal Stresses

A single run was performed incorporating stress caused by spinning of the disc. As the disc is rotating at 34.5Hz at its fastest (vehicle v_{max}), this value was used in the simulation. A maximum displacement was noted of 0.003mm which, when compared

with the equivalent displacement caused by thermal loads of about 1mm, is negligible. The inclusion of centrifugal effects was discounted.

4.3 'Reference' Disc Tests

4.3.1 'Inside-out' Design vs. Conventional Design

As has already been explained it was necessary to produce results for the current design of disc. This startpoint would form a reference against which the results of the proposed changes to the disc design could be evaluated. The two models were run with identical inputs and the results are shown in Figs 15, 16 & 17. The tables below provide a summary of the results, certain critical dimensions and distortions being selected and the 'inside-out' values compared with the conventional values. Fig 18 illustrates where measurements were taken on the models, the axes in the figure illustrating the directions in which displacements were taken to be positive. Coning angle was taken to be positive when the outside edge of the disc was displaced in the positive 'x' direction.

Quantity	Braking Face	'Inside-out'	Conventional	Difference (%)
Max. Face Temperature	1	849.2	849.5	0.0
	2	760.3	773.4	1.7
Mean Face Temperature	1	706.5	643.8	-9.7
	2	632.6	701.8	9.9
Shoulder Temperature	N/A	119.3	93.6	-27.5

Table 6. 'Reference' temperature summary¹.

The corresponding maximum face temperatures are very similar between the discs, these values being governed by pad pressure distribution as there is not sufficient time for the disc design to have any effect. The average face temperatures are also very close, but owing to the reversal in geometry, the 'inside-out' disc face #1 corresponds

¹ The 'Difference' value is derived by taking the absolute percentage difference and assigning a '-' where the conventional value is higher. It could be seen as a measure of the improvement of the 'inside-out' design over the conventional design.

to conventional face #2 and vice versa. The temperature at the shoulder of the conventional disc is much lower, owing to the shorter thermal path to the cool disc hub. It is worth noting the radial temperature distribution of the braking faces. There is a greater concentration of heat at the outside edge of face #1 owing to the poor pressure distribution of pad #1.

Quantity	Braking Face/Direction	'Inside-out'	Conventional	Difference (%)
Coning Angle (deg.)	1	0.48	-0.99	51.5
	2	0.53	-0.86	38.4
Edge Deflection (mm)	1	0.17	-1.33	87.2
	2	0.40	-1.10	63.6
Radial Expansion (mm)	1	1.21	0.94	-28.7
	2	0.98	1.27	22.8
Shoulder Displacement (mm)	X	0.04	0.07	42.9
	Y	0.50	0.49	-2.0

Table 7. 'Reference' displacement summary.

The most apparent difference in disc distortion is that the two discs cone in opposite directions. The conventional disc outer edge moves back toward the hub face whilst the 'inside-out' disc extends itself and moves away. There is also a reduction in coning angle in the 'inside-out' disc of between 40-50%. The spread (difference in coning angles of the two faces) calculated as a percentage is also similar between the two cases. The deflection at the outer edge corresponds to coning in the positive 'x' direction (see Fig 18.) whilst the 'y' direction (radial) displacement provides a measure of radial expansion. The expansion experienced by the two faces on each disc are again similar but on the opposite face. The radial displacement at the shoulder is practically identical in the two discs but there is significantly more positive axial displacement at this point on the conventional disc, even though the values themselves are very small.

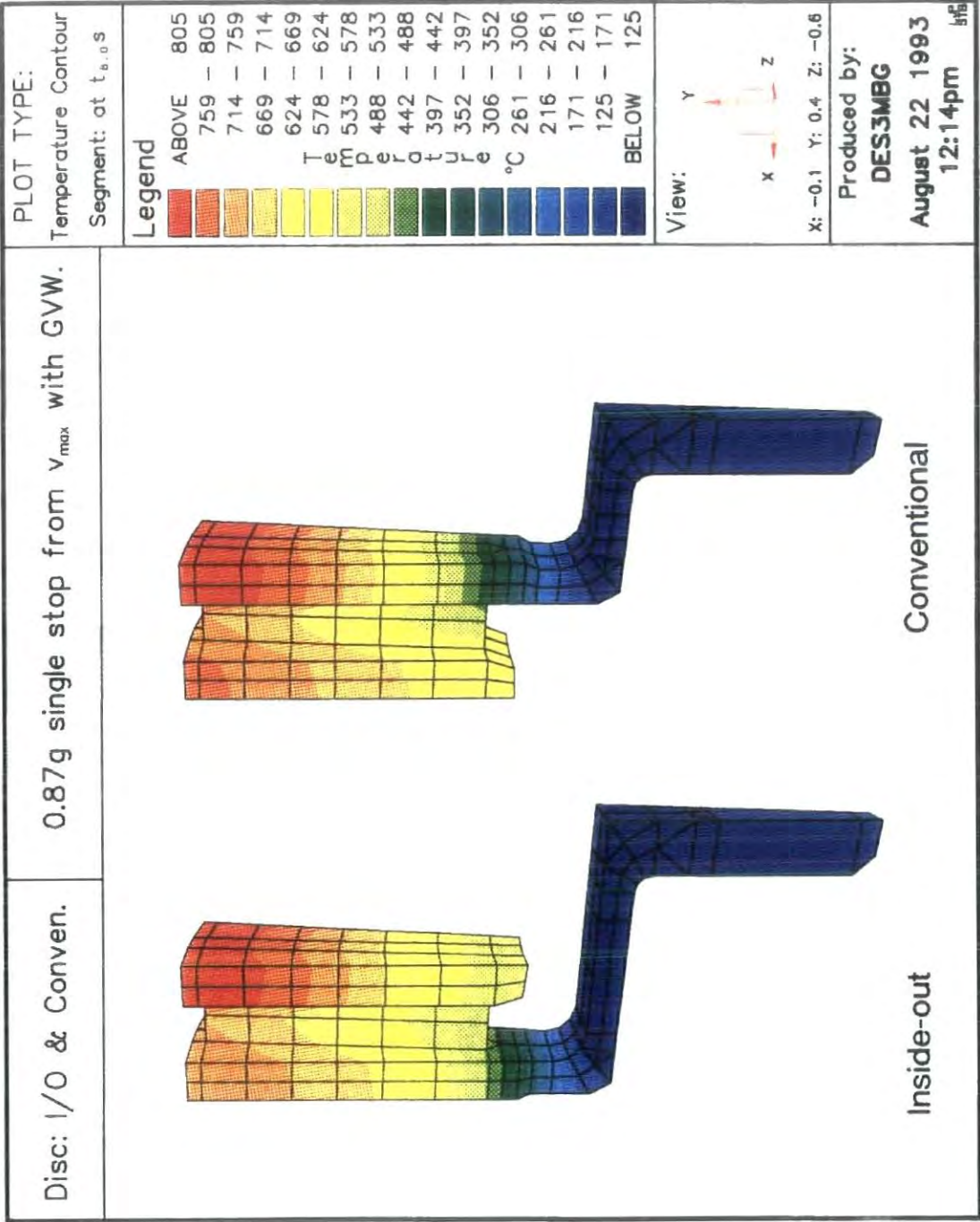


Fig 15. Temperature distributions - reference models .

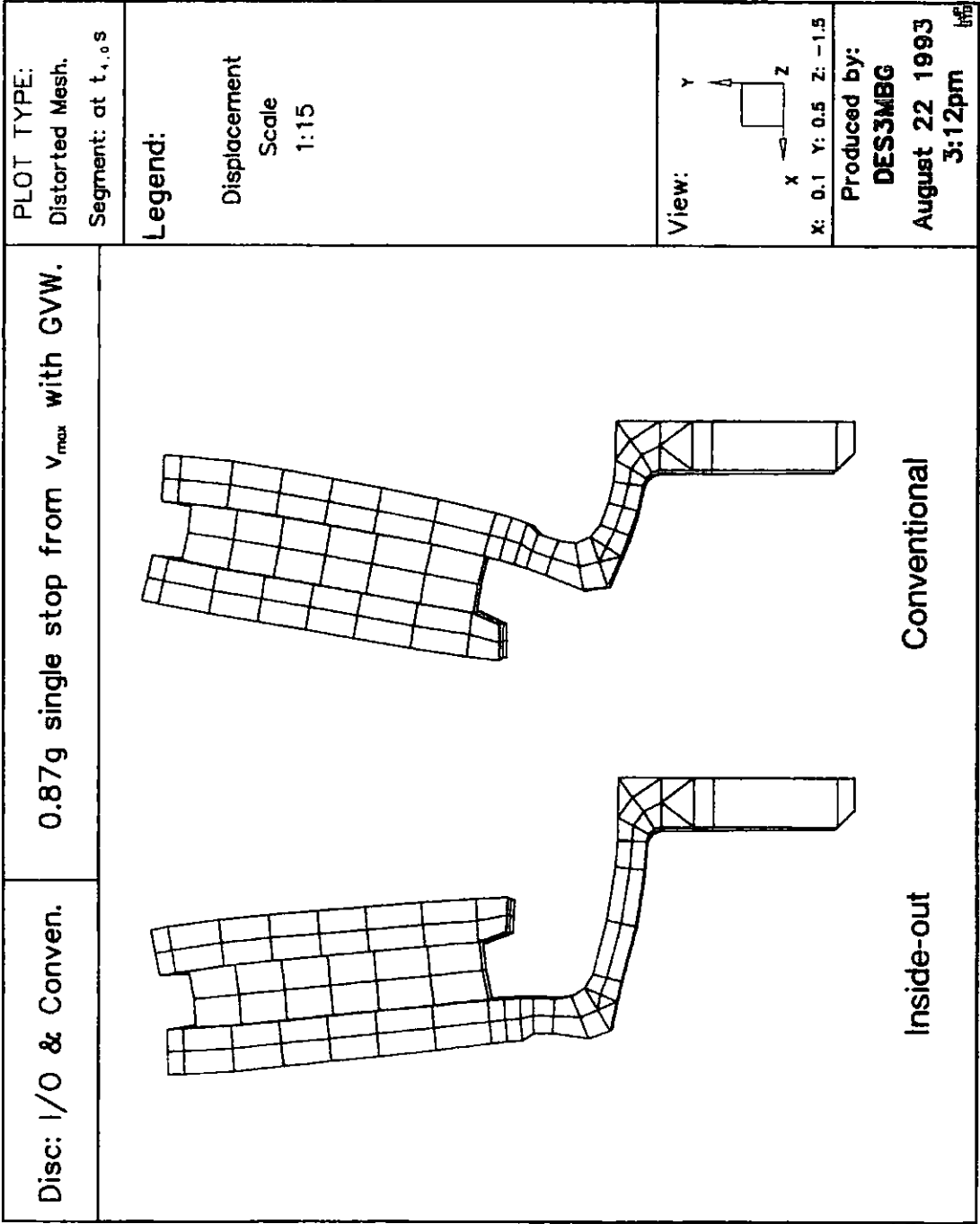


Fig 16. Displaced shape- reference models.

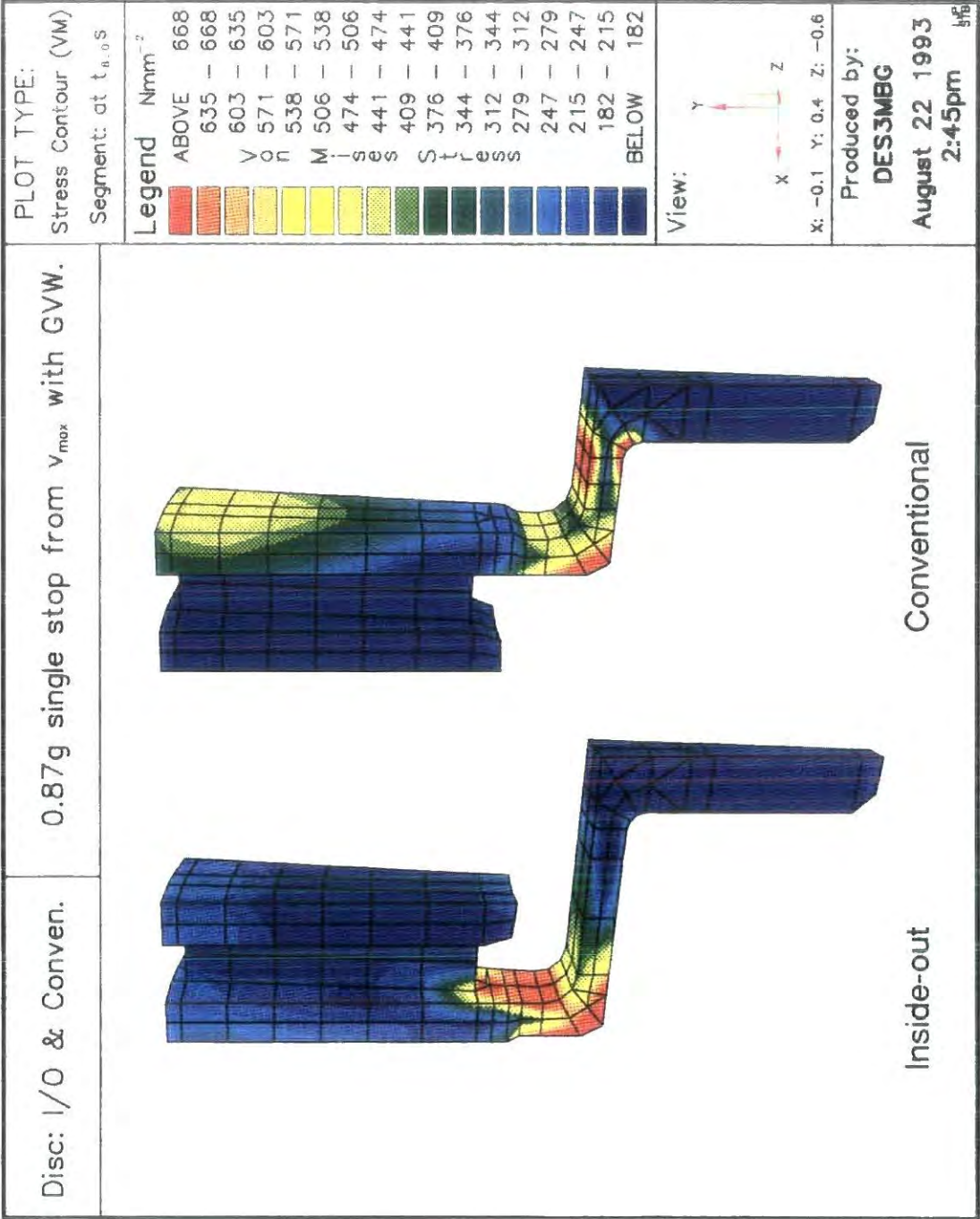


Fig 17. Stress distributions - reference models.

Location of Measurements

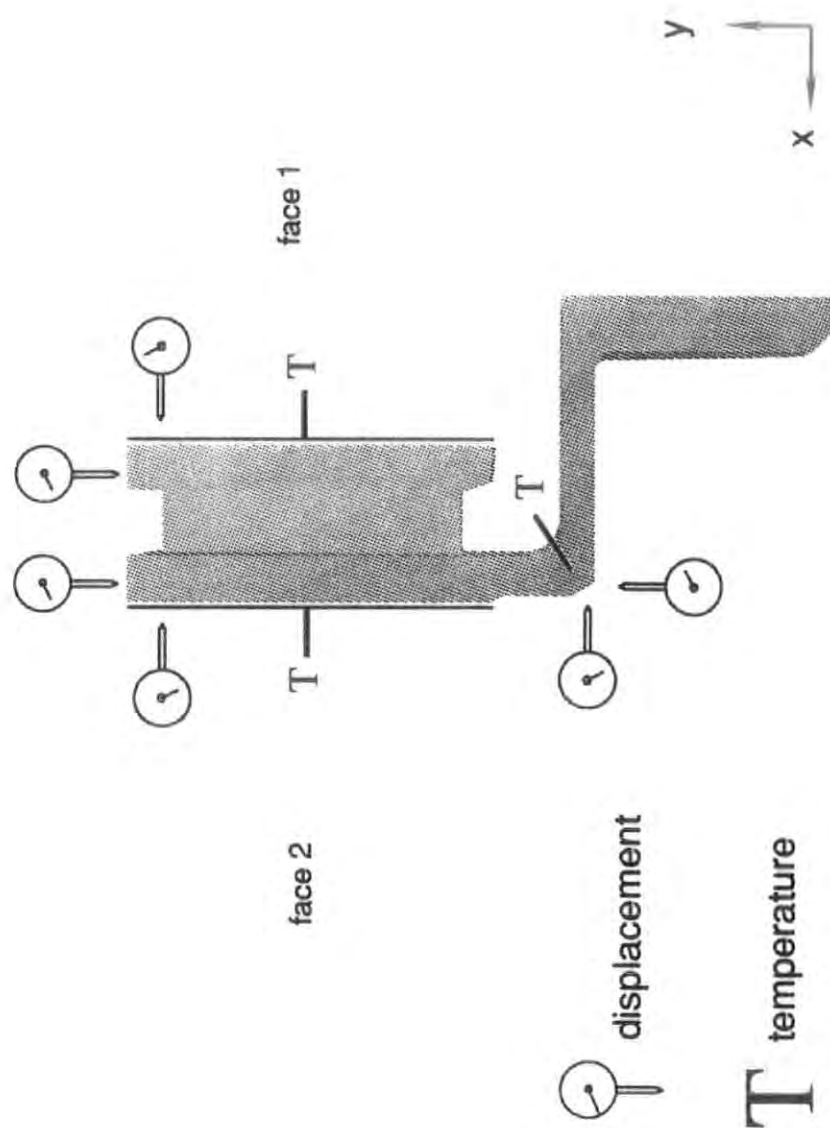


Fig 18. Location of measurements.

4.3.2 Solid Disc

A solid disc of the same mass and, therefore, thermal capacity as the other discs was run in the interest of comparison of performance. The results are summarised below, with Fig 19 containing the graphical output.

Quantity	Braking Face	Solid Disc
Max. Face Temperature	1	776.7
	2	757.3
Mean Face Temperature	1	663.2
	2	639.8
Shoulder Temperature	N/A	117.7

Table 8. Solid disc temperatures.

Quantity	Braking Face/Direction	Solid Disc
Coning Angle (deg.)	1	0.17
	2	0.25
Edge Deflection (mm)	1	0.04
	2	0.12
Radial Expansion (mm)	1	1.11
	2	1.05
Shoulder Displacement (mm)	X	0.04
	Y	0.55

Table 9. Solid disc displacements.

In the case of the solid disc, the maximum braking face temperatures are very similar to each other but generally equal to the equivalent value of each face #2 of the ventilated discs. This similarity across the faces of the disc is due to the concentration of the thermal capacity, giving the disc greater ability to absorb heat energy. The average temperatures are again very similar but in all cases a trend is evident where the face #1 temperature is slightly higher than the equivalent face #2 temperature, lending weight to the suggestion that this is caused by uneven pad thermal inputs.

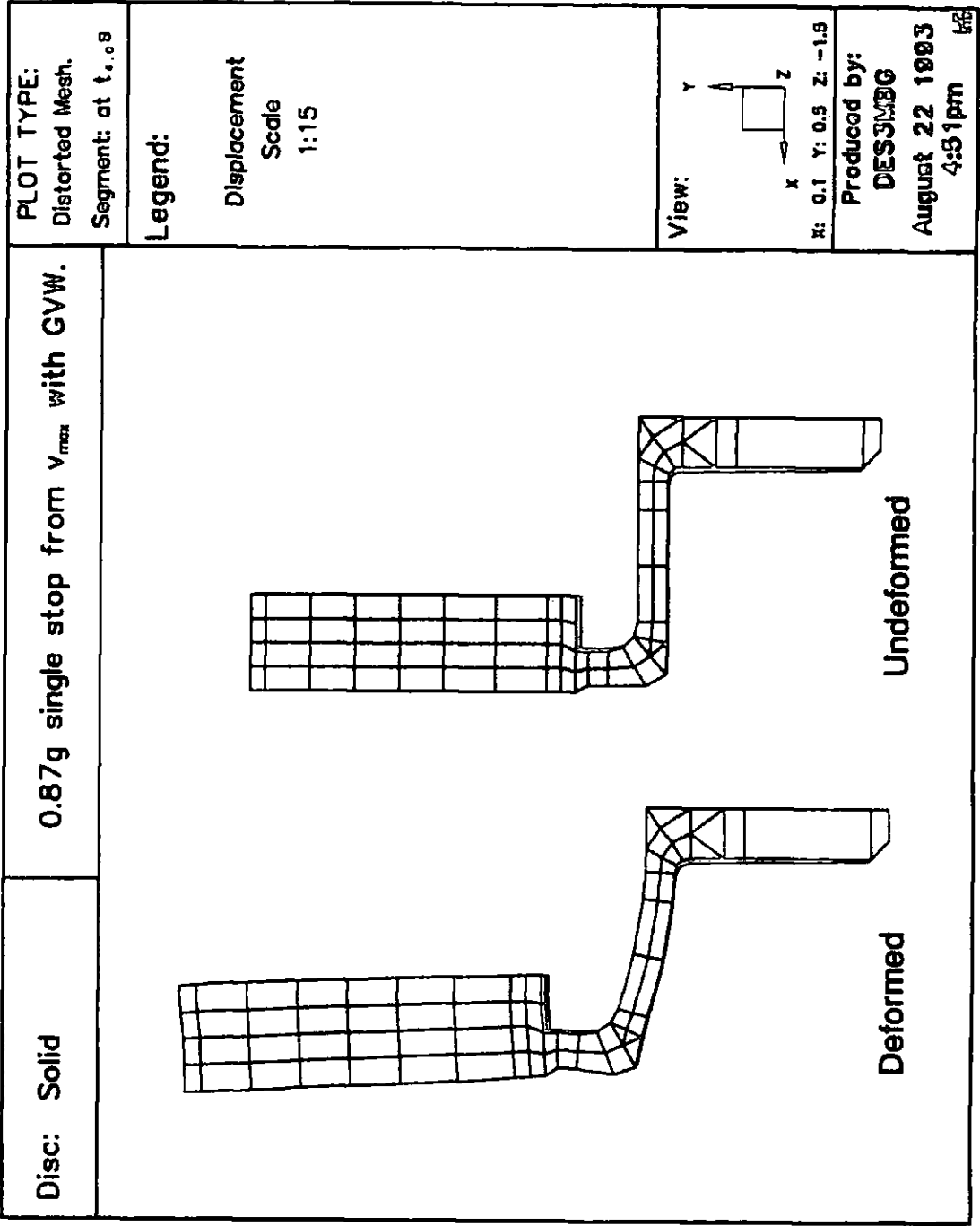


Fig 19. Solid disc mesh - undeformed and deformed.

The value of coning is much lower (50-60%) than the ventilated discs, but high spread of the two faces is encountered. This shows that this spread is independent of the ribs and is probably due to the faces expanding more toward the outside edge of the disc. The heat distribution is crucial and as it is known that face #1 receives a much greater thermal input near the outer edge this spreading may well be caused by the uneven expansion of face #1 with respect to face #2. The axial deflection of the disc is much lower (equivalent to the lower coning value observed) and there is approximately a 10% reduction in radial expansion. The heat is absorbed readily by the disc, resulting in lower distortions but will not be lost as easily and will build up quickly, the distortion advantages being lost as it fails to cool as quickly as the ventilated disc. Therefore this single stop test is particularly flattering to the solid disc. The shoulder displacement results are very similar to the 'inside-out' results.

4.4 Preliminary Hypothesis

The following are simple, initial observations about ventilated disc distortion behaviour that followed from the results presented above. They form the basic understanding of how the component responds under its working conditions. These preliminary observations formed the basis for the series of simulations, the main body of experimentation, described and presented in Section 5. The mechanism of coning was established for *solid* brake disc rotors by Abbas, Cubitt and Hooke [6] and some of the underlying distortion principles are shared by the two different types of disc.

Coning occurs because of the expansion of the braking faces (rotor cheeks) as they are subjected to rapid heating. The hub remains unaffected by this temperature rise in the time span of the test and therefore does not expand. Both faces try to expand equally, but one face is restrained, by its attachment to the disc wall. Fig 16 clearly shows this section of the disc wall being pulled outward by the expanding braking face (by about 0.5mm) and the wall appears to act as an encastre beam, deforming in a very similar way. This analogy is important and will be used later. The 'free' face is allowed to expand further than its restrained opposite and assumes a greater radial displacement with the result that faces are required to tilt to allow this differential of radial position of the two faces to be accommodated. This causes the disc to take the form of very shallow cone. This is analogous to a bi-metallic strip, one side expanding more than the other with a curve of the strip resulting. Using this explanation it is clear why coning occurs in opposite directions in the two differing types of discs with equal proportions.

In addition to the mechanical restraint of the disc wall, there is a further effect caused by the disc section only joining one of the rotor cheeks. The lower region of this rotor cheek is cooled more effectively by conduction and so does not expand as much. This effect is inherent in both designs of disc and is a further contributor to the coning mechanism.

If an improved design is to be generated it is necessary to examine the factors that govern the coning and its severity in more detail. The first of these is the effect of the disc wall.

It is necessary to investigate this characteristic, examining the effect of offering either more or less resistance to the expanding rotor cheeks. Offering less resistance would, according to the above theory, decrease the amount of coning, the disc faces being allowed more even, radial expansion. However reducing the stiffness of this part of the disc section could also severely reduce the mechanical strength of the disc. It can be seen from the displacement results that the shoulders of the two discs are both pulled out to the same radial displacement. This suggests that the reaction of the expanding rotor cheeks far exceeds the resisting force supplied by the wall as the disc walls are of substantially different lengths (and therefore using the beam analogy, stiffness).

It could therefore be argued that it is not mechanical resistance that causes the coning because, if the expansion force so exceeds the wall restraining force that such greatly differing wall stiffnesses make little difference, then it must be the localised cooling effect at the inner edge of the braking face attached to the wall. This face would then not be expanding as much as its unsupported counterpart but sufficiently to impart the irresistible outward force to the base of the disc wall. The effects of these need to be examined in more detail, starting with an analysis of the effect of disc wall stiffness on coning.

The 'inside-out' disc was designed to counteract thermal distortion and it is inherently better for a number of reasons. Firstly as already discussed the disc wall length is longer and it is possible that this may contribute to a reduced resistance to expansion. Secondly the shoulder temperature is higher and the differential of temperature across the supported disc face will be reduced. Thirdly, the longer wall depth means that for a given radial displacement at the shoulder there will be a smaller deviation of the angle of the shoulder. This is illustrated in Fig 20.

Differential in Shoulder Angle

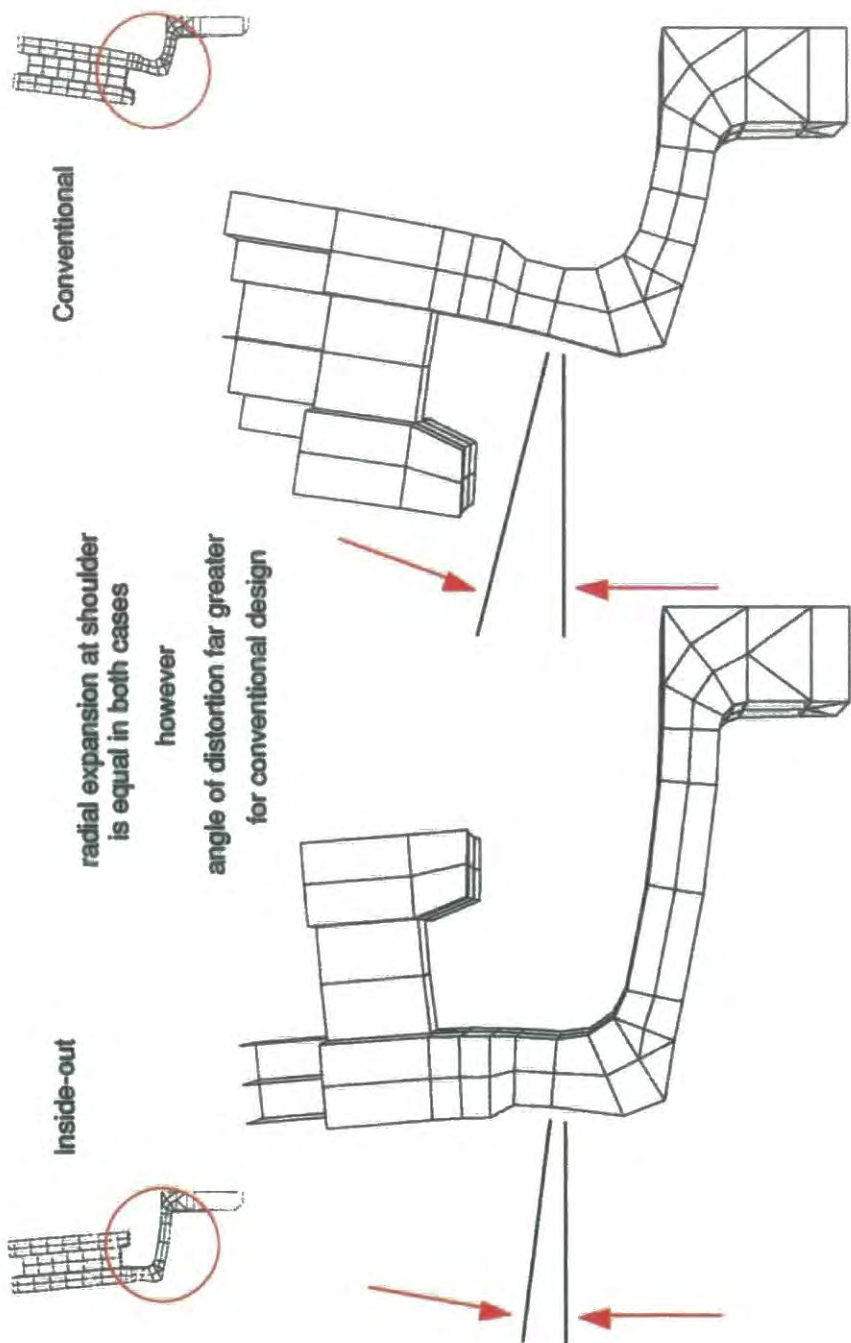


Fig 20. Effect of disc depth on shoulder distortion angle.

The effect of this relatively severe angle change in a conventional disc would be to act in support of the uneven expansion in creating bad coning. In the 'inside-out' disc the uneven expansion of the faces is reversed, producing a moment at the base of the braking face that tends to counteract the effect of the shoulder angle, producing a total rotor cheek angle closer to 0°.

Previous papers (and Ford tests) have all shown that 'inside-out' discs have a reduced susceptibility to coning. However their cooling performance is not equal to that of conventional discs and this could be caused by the greater thermal path between the braking faces and hub or the more tortuous route that needs to be taken for air to move through the cooling passages of an 'inside-out' disc. In this case it could depend on what the priority characteristics are of the disc being designed, as to whether it is designed with an 'inside-out' or conventional section.

The aim of this research is to produce an improved design and so it is necessary to incorporate the desirable characteristics of both discs into an alternate design.

It appears that there are two ways of increasing disc performance with respect to coning. Firstly, reduce the resistance to expansion that causes coning and secondly, to remove the asymmetrical nature of the two faces that is the root of coning.

With the 'inside-out' design the coning effects counteract the shoulder angle change, but this results in considerable deformation of the shoulder and high stresses within it. High stresses are undesirable as cracking can be caused, particularly during the inherently cyclic nature of disc use, and local plastic deformation can occur. Both of these are highly undesirable and must be avoided. Therefore a design must be found where these stresses are kept low. Disc thermal stresses occur mainly in the shoulder and it was decided to conduct trials into the design of this shoulder with the aim of reducing the stresses induced and the resistance offered to rotor cheek expansion.

It was also considered feasible that if the disc faces were designed in such a way as to expand unequally themselves, then this could be used to cancel out the effect of one of them being restrained. By doing this the minimum of additional stresses would be created because a solution would have been attained by allowing the disc to expand naturally (without coning) and the high stresses induced by trying to restrain an

effectively irresistible force avoided. It was felt that this approach of allowing the disc to 'behave as it wishes' whilst designing it so it naturally deforms radially but without coning would produce a disc with lower stress levels and less mass (less material would be required as a lesser, not greater, restraining force is required). The decision was made to devote some of the testing to achieving this end.

It was also decided, in contrast to the above, to try and reduce coning by building up the shoulder section in an 'inside-out' disc to try and reduce the opening out of the right angle at the shoulder, therefore reducing coning and possibly lowering stresses.

Other deformation effects can be noted. The rotor cheeks at the outer edge of the disc have a tendency to curl over towards each other. This is because they are unsupported, the ribs not extending to the disc diameter. Some testing would be performed to examine and rectify this phenomenon as it clearly results in a loss of contact pressure, an altered pressure distribution and reduced braking effectiveness. It does appear that initial poor pad distribution may be a contributory factor because owing to the pressure concentration on the outer edge of pad #1 and greater disc speed at this point there will be an increased heat input into this unsupported part of the braking face.

Rib thickness would also be investigated to try and draw heat away from the braking faces more effectively, resulting in less thermal expansion.

Some spread of the rotor cheeks was also observed (greater distance between rotor faces at the outside edge of the disc than the inside) and this was thought possibly to be caused by rib expansion, greater towards the outside because of increasing heat input. However it is also possible that it is the faces themselves whose thickness is increasing unevenly and again poor pad pressure distribution, particularly that of pad #1, is a causal factor.

CHAPTER 5

RESULTS

The majority of the simulations performed followed on from its predecessor in terms of the subject of the testing and so the following Section detailing the main body of experimentation is written in a similar manner. For each simulation run, the reasons and details of the test, the results and the interpretation of these results are contained together. There were three main stages of experimentation. The first used segment models and the single stop test, the second (also employing segment models) demonstrated cooling and the third stage utilised full disc models, in the investigation of both thermal and mechanical waving.

5.1 Disc depth

5.1.1 Aims and Procedure

It was decided to vary the depth of the disc, effectively changing the disc wall stiffness, and observe the changes in coning obtained. Five identical simulation runs were performed with the disc depth being varied from 43mm to 27mm. An 'inside-out' design of disc was used.

5.1.2 Results

There was no change to the temperatures on the disc and the displacement summary results are given in Table 10. There was a 10% decrease in the radial displacement at the shoulder for A c.f. E (for the same two models there is a 33% difference in axial displacement). Coning increases as the depth of the disc decreases, and a graph illustrating this is shown in Fig 21.

Quantity	Braking Face/Direction	Case A - 43mm	Case E - 27mm	Difference (%)
Coning Angle (deg.)	1	0.474	0.543	12.7
	2	0.523	0.565	7.4
Edge Deflection (mm)	1	0.157	0.079	-98.7
	2	0.385	0.307	-25.4
Radial Expansion (mm)	1	1.212	1.195	-1.4
	2	0.978	0.934	-4.7
Shoulder Displacement (mm)	X	0.036	0.054	33.3
	Y	0.502	0.453	-10.8

Table 10. Displacement summary - disc depth variation.

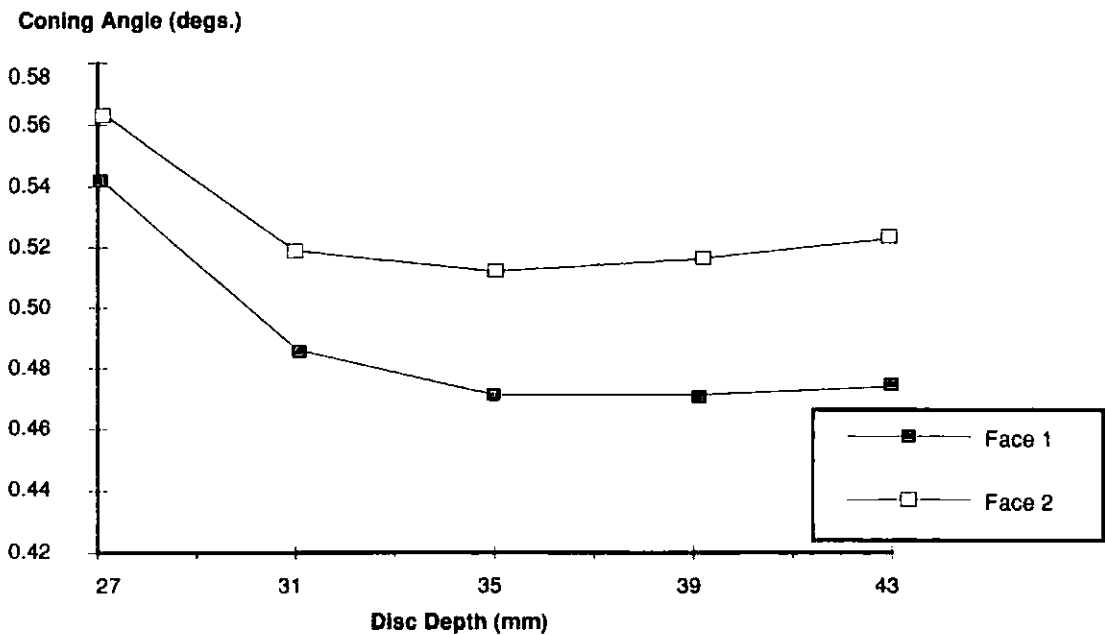


Fig 21. Variation of coning with disc depth.

5.1.3 Interpretation

It can be concluded that wall section stiffness affects the coning angle of the disc because of the increased resistance to rotor cheek expansion. Shoulder temperatures do not change, the heat not having time to permeate along the altered length of changed wall depth, and therefore localised conductive cooling at the base of the supported face is unlikely to be the major resistance to expansion, causing coning. The resistance is such that in the 'inside-out' disc, as the shoulder angle becomes more severe with less depth (acting to diminish the coning angle), the restraining effect on the rotor cheeks counteracting it becomes proportionally greater, overpowering the effect of the shoulder angle and *more* severe coning occurs. This might explain the marginal improvement in coning between disc depths of 43mm and 35mm. It is possible that in a conventional disc, lengthening the disc wall would serve a double benefit, firstly providing less expansion resistance and secondly reducing the change in shoulder angle with a given radial displacement. The increased disc depth is also likely to reduce disc mechanical stiffness and this would require further examination.

5.2 Rib Thickness

5.2.1 Aims and Procedure

The width of the ventilating ribs was varied between 6mm & 10mm across five simulations. It was hoped that a reduction in face temperatures, and therefore expansion, could result from an increase in rib thickness. The current width of rib is 5mm.

5.2.2 Results

The thicker ribs showed marked improvements. The average face temperatures for the two braking faces achieved a 16-18% reduction in temperature and approximately a 7% reduction in maximum temperature values. A 29% improvement in coning of face #1 and 15% improvement in face #2 resulted from these temperatures reductions. These improvements far exceed the percentage increase in Thermal Capacity. However the spread of the two faces practically doubles.

Quantity	Braking Face	Case A - 6mm	Case E - 10mm	Difference (%)
Max. Face Temperature	1	833.8	768.6	7.8
	2	747.5	693.6	7.2
Mean Face Temperature	1	690.2	569.3	17.5
	2	615.4	514.5	16.4
Shoulder Temperature	N/A	115.0	94.7	17.7

Table 11. Temperature summary - rib thickness variation.

Quantity	Braking Face/Direction	Case A - 6mm	Case E - 10mm	Difference (%)
Coning Angle (deg.)	1	0.47	0.34	29.1
	2	0.53	0.45	14.9
Edge Deflection (mm)	1	0.18	0.14	25.0
	2	0.41	0.35	14.0
Radial Expansion (mm)	1	1.18	0.97	17.7
	2	0.95	0.79	16.8
Shoulder Displacement (mm)	X	0.04	0.03	17.1
	Y	0.49	0.41	16.0

Table 12. Displacement summary - rib thickness variation.

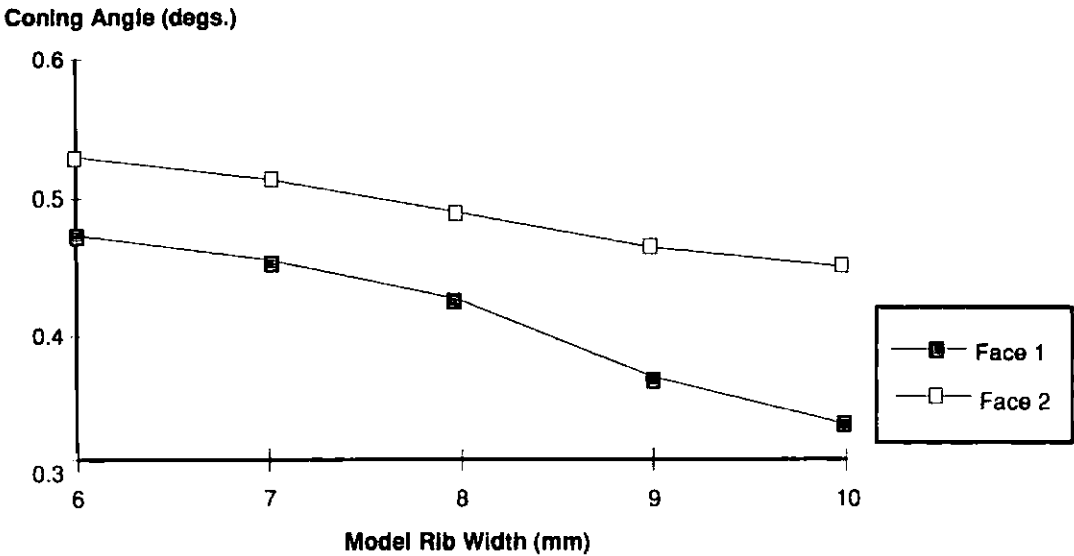


Fig 22. Variation of coning with rib thickness.

5.2.3 Interpretation

The lower average face temperatures result from the greater bulk of ribs absorbing the heat energy from the rotor faces. The maximum temperatures are little affected probably because they occur at the outer edge, owing to the poor pad pressure distribution and the fact that the disc ribs do not quite extend to the edge. As a result a significant reduction in coning is achieved but the increase in rotor cheek spread may be due to greater rib temperature and therefore expansion along its length. In all of this there is however a weight penalty although the increase in weight of roughly 10% does not outweigh the reduced distortion. Disc cooling may also be affected by the reduction in size of ventilating passages but this unlikely to be very severe.

5.3 Uneven Face Thickness - 'Inside-out'

5.3.1 Aims and Procedure

As briefly mentioned in Section 4.4 it was decided to try and design the disc rotor faces so that they would naturally expand unevenly and cone, but in an opposite nature to that which would normally occur. In this way the two effects would oppose each other and a reduction in coning might be obtained. To this end five disc models were developed in which the thickness of the unsupported face (#1) of the 'inside-out' disc varied between 4mm and 12mm (the current thickness is 7mm). In varying this dimension it was hoped that a variation in the radial expansion of the face could be achieved. In theory, a thicker face would absorb the given heat flux input, resulting in a reduced temperature rise of the face and a corresponding reduced expansion. If the face that forces itself past the other to cause coning is thickened then it was hoped that unrestrained thicker face would naturally expand the same amount as the thinner face (at a higher temperature but restrained by the disc wall). A suitable ratio of face thicknesses would have to be found.

5.3.2 Results

The model distortions are shown in Fig 24 and the summary table given below. The results of the '10mm' test are compared with those for the '6mm', close to the current design of 7mm. A 30% decrease in the average face #1 temperatures was achieved as

expected, a 4% decrease being experienced by the restrained face. Very little difference was found in the shoulder temperature. As far as displacements were concerned a reduction of the coning from 0.48° and 0.53° for face #1 and #2 respectively to -0.068° and -0.009° was observed. The graph below shows the average face coning results.

Quantity	Braking Face	Case B - 6mm	Case D - 10mm	Difference (%)
Max. Face Temperature	1	958.8	646.3	32.6
	2	767.2	746.6	2.7
Mean Face Temperature	1	788.6	548.8	30.4
	2	641.3	615.1	4.1
Shoulder Temperature	N/A	119.7	118.8	0.7

Table 13. Temperature summary - uneven face thickness (i/o).

Quantity	Braking Face/Direction	Case B - 6mm	Case D - 10mm	Difference (%)
Coning Angle (deg.)	1	0.77	-0.07	91.1
	2	0.81	-0.01	98.9
Edge Deflection (mm)	1	0.46	-0.37	20.2
	2	0.69	-0.16	77.2
Radial Expansion (mm)	1	1.35	0.92	32.3
	2	0.98	0.98	-0.2
Shoulder Displacement (mm)	X	0.03	0.04	-37.5
	Y	0.51	0.48	6.3

Table 14. Displacement summary - uneven face thickness (i/o).

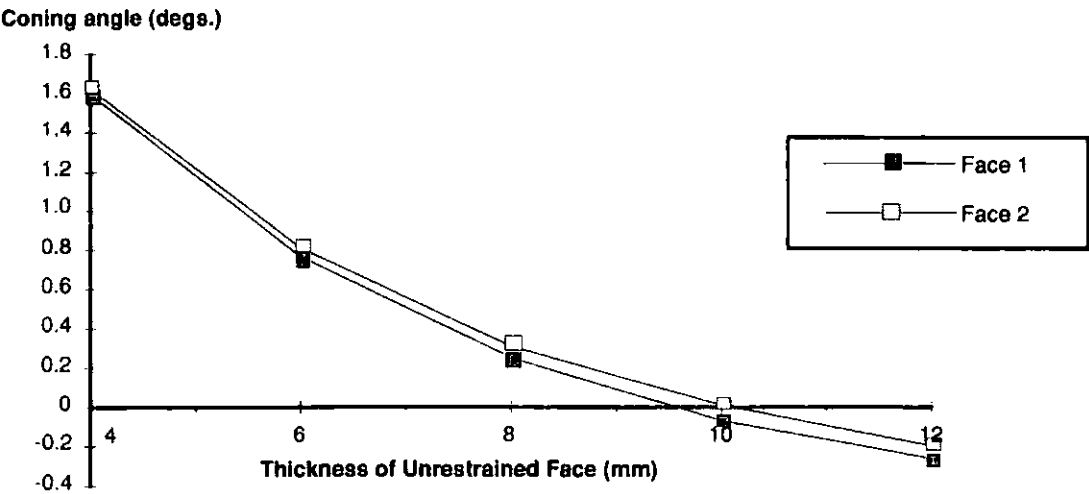


Fig 23. Variation of coning with face #1 thickness (i/o).

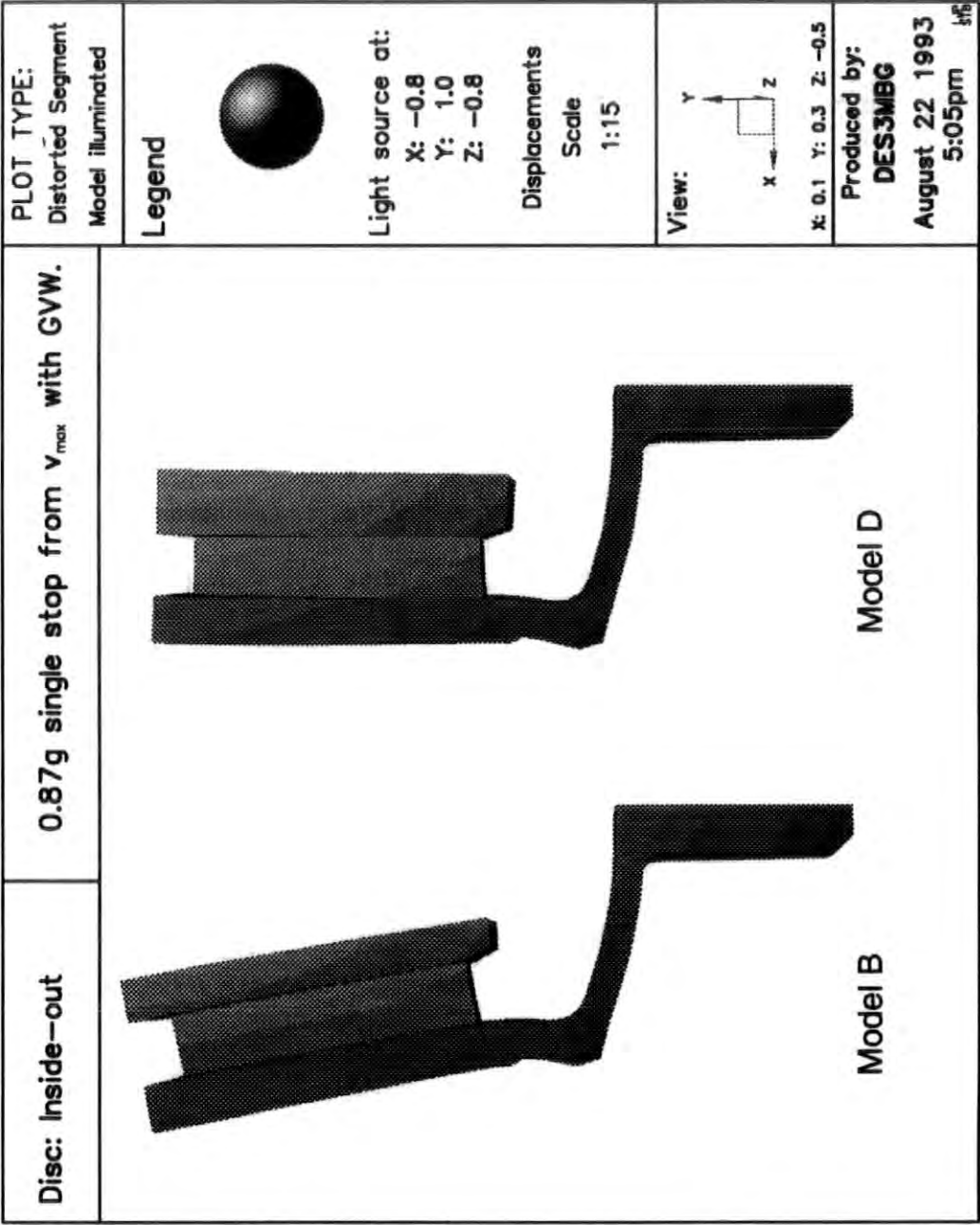


Fig 24. Displaced shape - uneven face thickness (i/o).

5.3.3 Interpretation

The range of coning achieved shows that this method can be used to good effect. However there are possible drawbacks. Firstly, there is naturally a mass increase associated with this method (approximately 12.5%) and therefore should only be used if distortion performance is of a higher priority than the resulting increase in mass. Secondly the thicker face is not likely to cool as effectively (although if combined with thicker ribs as already shown this disadvantage may be reduced) and poorer performance may be shown over multi-stop braking tests. The response of this design of disc in near steady-state conditions is important and needs to be examined further. In addition, the effect of uneven face thickness on mechanical stiffness of the disc needs to be ascertained. However there appears to be potential for this method of design in the reduction of coning.

5.4 Uneven Face Thickness - Conventional

5.4.1 Aims and Procedure

As the simulations described above were performed on an 'inside-out' design it was decided to try and reproduce the effect on conventional discs. The same variation of face thickness was used, this time on face #2.

5.4.2 Results

The results are summarised below. A practically identical reduction in face temperatures was obtained and an almost as impressive improvement in coning as with the 'inside-out' disc. Between 85% and 100% reduction was observed, face #1 values reducing from -0.99° to -0.147° and face #2 from -0.858° to -0.003° .

5.4.3 Interpretation

To achieve this same reduction of coning a 12mm face was required because of the greater amount of coning in the original design to be overcome. An increased penalty therefore exists but the improved cooling of conventional designs of discs is retained, important if the cooling of a thicker face is to be aided.

Quantity	Braking Face	Case B - 6mm	Case E - 12mm	Difference (%)
Max. Face Temperature	1	855.8	832.2	2.8
	2	873.2	521.0	40.3
Mean Face Temperature	1	652.5	619.1	5.1
	2	783.4	486.1	38.0
Shoulder Temperature	N/A	93.9	92.6	1.4

Table 15. Temperature summary - uneven face thickness (conv.).

Quantity	Braking Face/Direction	Case B - 6mm	Case E - 12mm	Difference (%)
Coning Angle (deg.)	1	-1.31	-0.15	88.8
	2	-1.19	0.00	99.7
Edge Deflection (mm)	1	-1.68	-0.41	75.4
	2	-1.44	-0.22	85.0
Radial Expansion (mm)	1	0.94	0.93	1.3
	2	1.42	0.82	42.2
Shoulder Displacement (mm)	X	0.07	0.05	28.2
	Y	0.51	0.43	14.8

Table 16. Displacement summary - uneven face thickness (conv.).

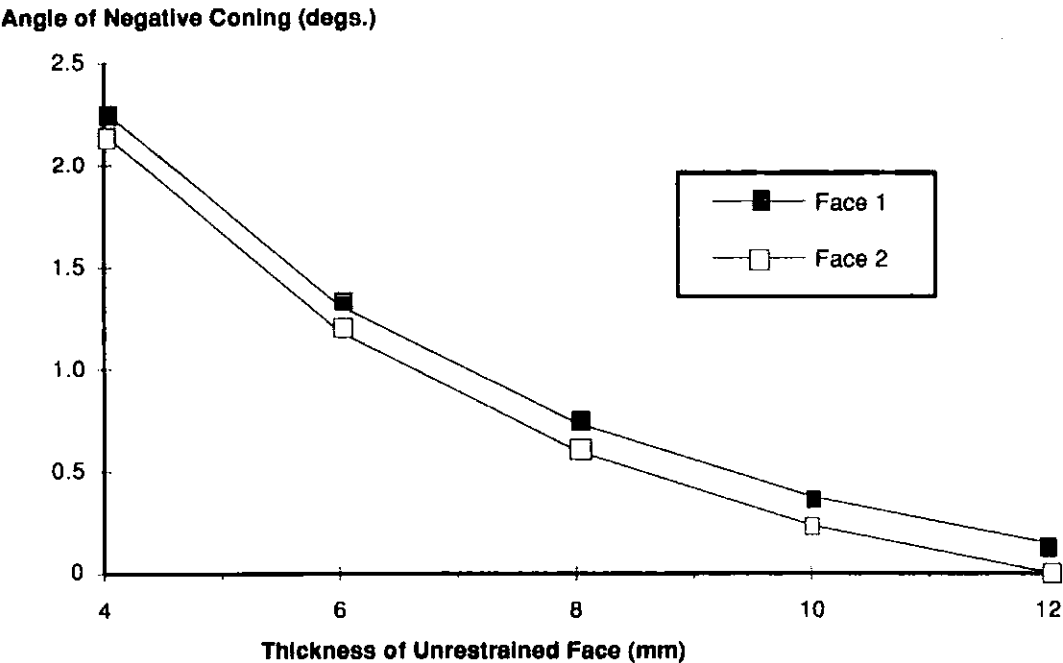


Fig 25. Variation of coning with face #1 thickness (conv.).

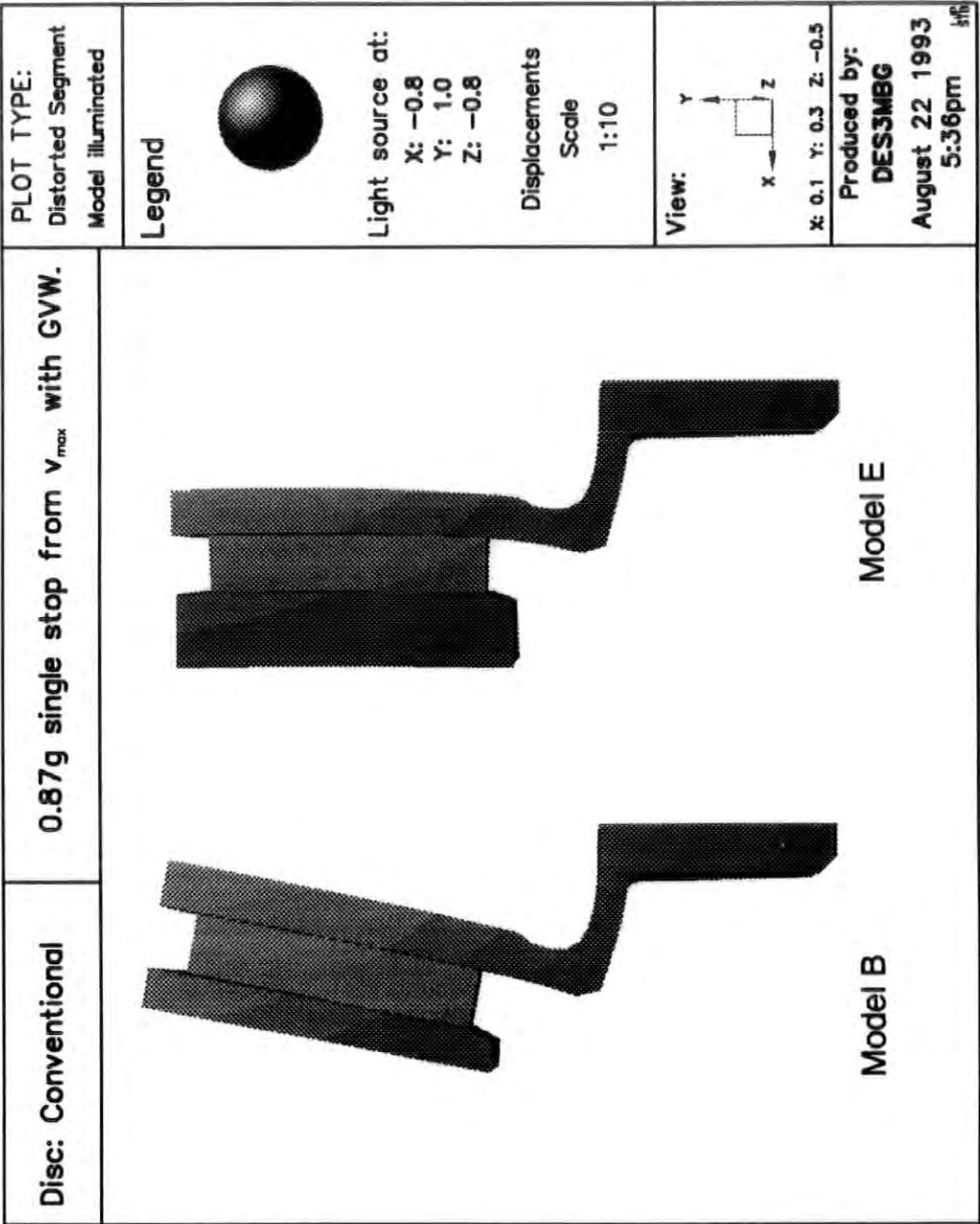


Fig 26. Displaced shape - uneven face thickness (conv.).

5.5 Undercut Variation

5.5.1 Aims and Procedure

Another way of decreasing the resistance to face expansion is to introduce and increase the depth of an undercut on the conventional design of disc. The undercut took the form of a groove machined just inside braking face #1. In this way there would be a tendency for the shoulder angle to 'open up' more, reducing the resistance offered by the wall section to face #1 expansion. It was decided to model the effect of this undercut on coning and five models were produced, the undercut depth varying in even steps from 1mm to 3mm.

5.5.2 Results

A 21% decrease in coning was observed across the range of the models (this figure does not compare directly with the conventional disc reference value) with a slight decrease in average face temperatures.

Quantity	Braking Face/Direction	Case A - 1.0mm	Case E - 3.0mm	Difference (%)
Coning Angle (deg.)	1	-0.84	-0.78	7.1
	2	-0.80	-0.76	5.2
Edge Deflection (mm)	1	-1.16	-1.07	7.3
	2	-0.95	-0.87	8.9
Radial Expansion (mm)	1	0.95	0.95	-0.2
	2	1.29	1.28	1.2
Shoulder Displacement (mm)	X	N/A	N/A	N/A
	Y	N/A	N/A	N/A

Table 17. Displacement summary - undercut variation.

5.5.3 Interpretation

Because of the slightly altered restraint conditions, the reduced resistance to expansion caused by the opening of the undercut results in less coning. This effect would be more marked in a solid brake disc, the dimensions of the undercut playing a more

crucial role in the coning response of the disc. In addition the implementation of this design feature does not carry with it a disc weight penalty.

5.6 Number of Ribs

5.6.1 Aims and Procedure

It was decided to examine the effect that a variation of the number of ribs would have. The model segment size was varied, the rib width being varied proportionally to maintain the Thermal Capacity of the disc. Four models were developed for comparison, total rib numbers varying from 25 to 55 (corresponding to a segment angle change of 14.4° to 6.55°).

5.6.2 Results

Very little difference was observed between the models. Only 1.3% variation of face temperatures was obtained across the range and even less displacement change.

5.6.3 Interpretation

Little advantage was evident in the variation of rib numbers. An increased number would spread the heat more evenly, reducing any rippling of the disc rotor cheek between ribs, although this is not thought to have a large effect on disc distortion performance. A greater number of ribs does mean a larger surface area from which to lose heat and the effect of this on the cooling performance of the disc could not be modelled for reasons explained earlier. An improvement might be expected, although there is naturally a change in air passage flow conditions.

5.7 Pad Material

5.7.1 Aims and Procedure

The importance of the pad pressure distribution has been mentioned frequently in this report. This is very fundamental to the behaviour of the disc, and even though the project brief is to examine disc design it was decided that the root of many of the

problems may be in the pad/caliper design. The pressure distributions are very different as has already been shown and this difference is due to the different contact areas on the backplates of the pads. Pad #1 has a very undesirable concentration of pressure near the outside edge of the disc, where disc surface speed is also highest, and as close to uniform as possible pad pressure distribution is required for favourable disc response [24]. Caliper design is too complex to model quickly but as a pad model was existing already it was decided to utilise it to produce a small test on the importance of pad design in achieving the desirable even pad pressure distribution.

Harding and Wintle [12] studied this for pads with high aspect ratios and showed that it is the combination of friction material compressibility, pad load and load pattern in addition to backplate thickness that affect the performance of the pad. It was decided to produce a comparison of two of these effects in this particular instance.

The first characteristic was chosen to be the compressibility of the friction pad itself. Two models were defined, one with a higher than current value of Young's Modulus for the friction material and one with a lower value (these material values and others used in the pad models are given in Appendix I). A comparison was made of the stress distributions.

5.7.2 Results

The following graphs show the pad/disc pressure over a radial path on the braking face at the centre of the pad. Fig 29 shows the pad face pressure distributions for the high compressibility ('softer') pad material and the current pad. The same scale is used in each case. As is clear from the figures, the softer pad material produces a much more uniform distribution, compensating in many ways for the limitations of the caliper design.

5.7.3 Interpretation

It is shown to be desirable to use a softer pad material for a favourable pressure distribution, as previously demonstrated by Harding and Wintle [12]. However there are often difficulties with using a softer pad, and creating a material which is soft and yet still has a very high wear resistance and the capability to operate under the very high pressures and temperatures of the pad/disc interface without degrading is difficult. Softer pad material may also affect the brake response and feel.

Contact Pressure over Radial Pad Centreline - face 1

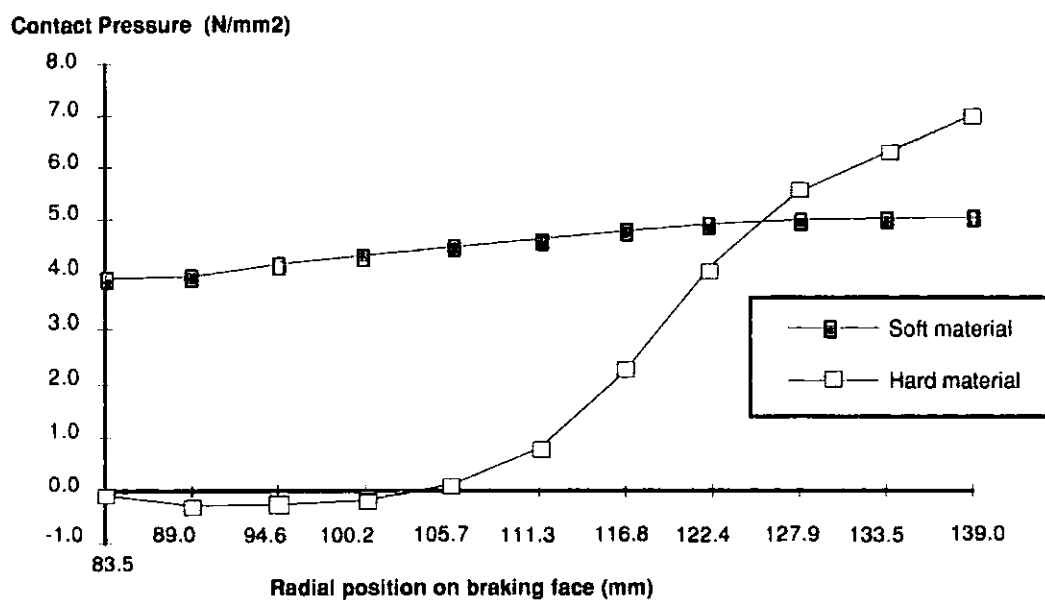


Fig 27. Contact pressures for differing materials - face #1.

Contact Pressure over Radial Pad Centreline - face 2

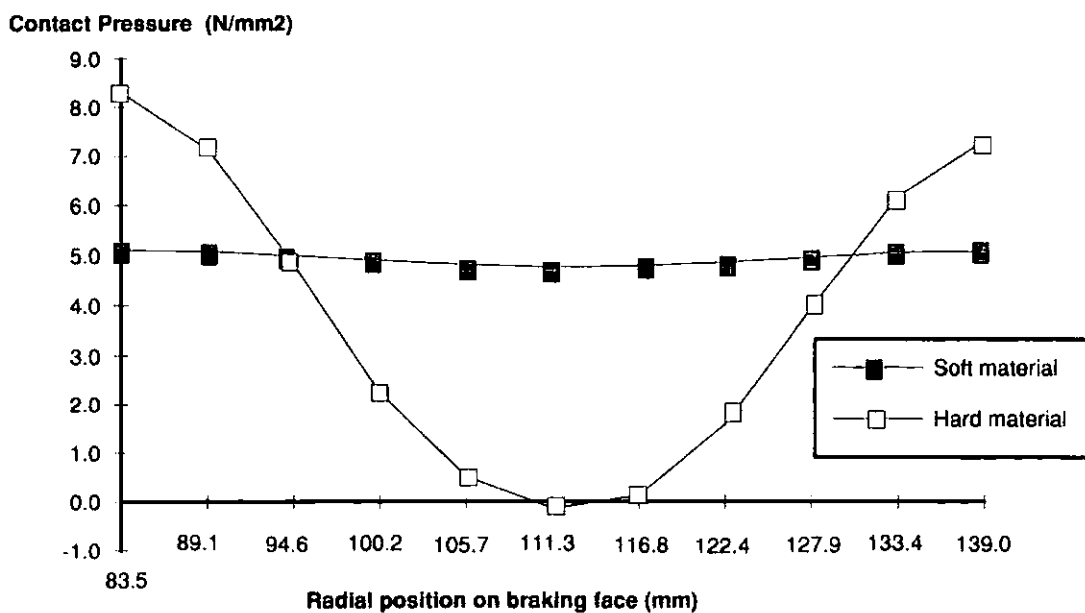


Fig 28. Contact pressures for differing materials - face #2.

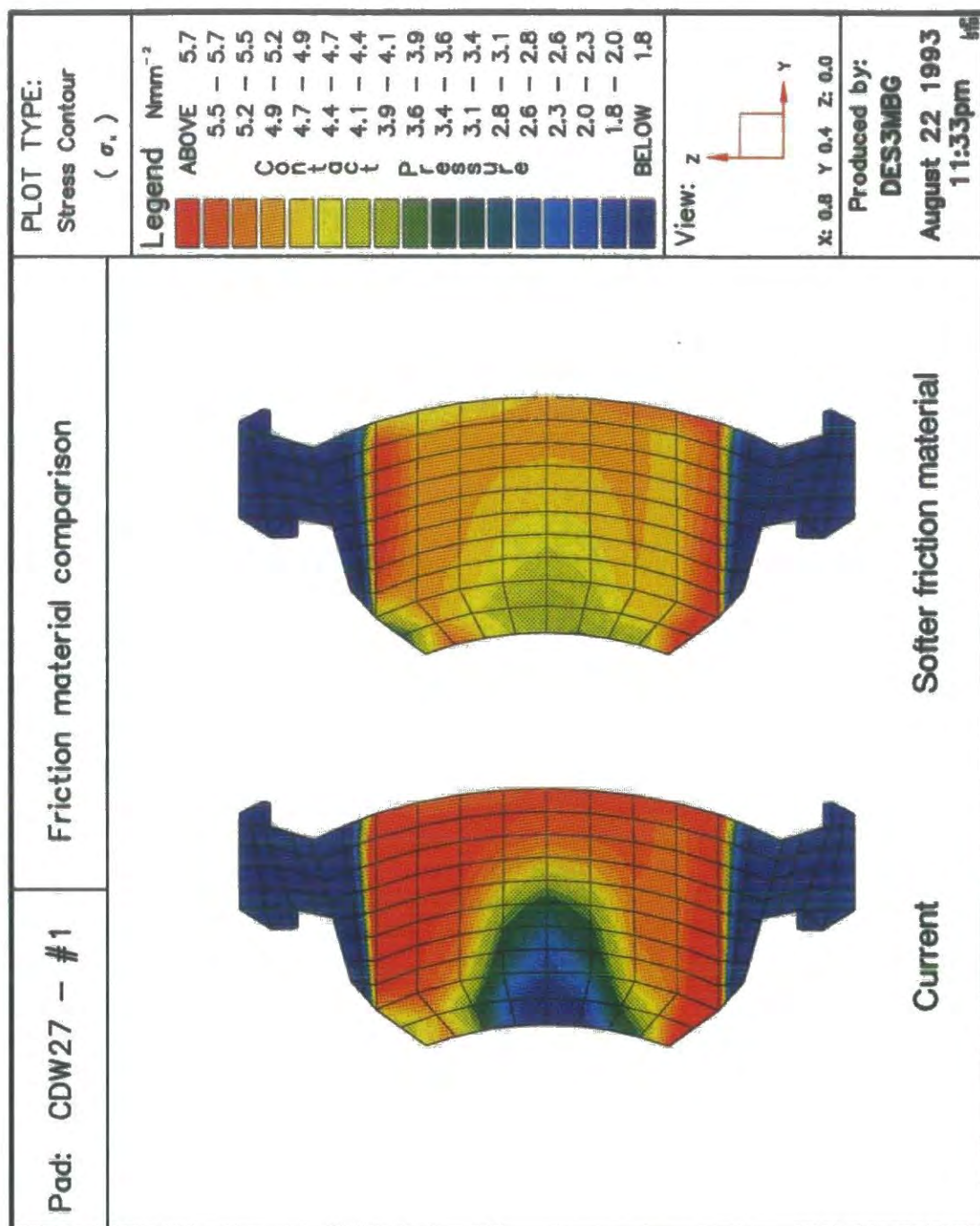


Fig 29. Pad/disc contact pressure distributions - material compressibility.

5.8 Pad Backplate Thickness

5.8.1 Aims and Procedure

Another method of producing a more uniform pressure distribution is to examine the backplate thickness.

Four principal models were constructed, backplate thicknesses varying from 5.5mm to 14.5mm. The current backplate thickness is 5.5mm and the models were identical in all other aspects. Simulations were then run using the standard loading conditions and loading patterns for both pads used.

5.8.2 Results

As expected the thicker backplates produced a much more uniform distribution. The results are shown in Figs 30,31 & 32. Fig 32 compares the thicker backplate with the current pad design.

5.8.3 Interpretation

Thicker backplates are naturally able to reduce the variation of pressure. A significant reduction was achieved using a 14.5mm backplate but pad backplates are usually designed to keep material costs low. However a thicker backplate could be used where a softer friction pad material is not available. This simple and relatively inexpensive alteration to pad design could produce much more favourable disc response by virtue of the fact that the pad design has much influence on disc distortion by supplying a non-uniform pressure. This revision could be used where the re-design of the caliper is undesirable through cost, packaging and even manufacturing limitations (the pad #1 contact faces of the caliper are shaped so that a boring tool can be passed between them to machine the cylinder bore).

An additional simulation was performed where the friction face of the pad was displaced a fixed amount (the backplate being held rigid) to see what the corresponding contact pressure rise would be. In this way the magnitude of the effect that runout might have on the contact pressure could be assessed. The friction face was displaced by 0.035mm, the maximum value that would occur if the pad were

considered rigid and a typical disc runout of 0.07mm existed. The contact pressure rose to 5 times its normal value. Although the rigid pad is not an accurate assumption this did demonstrate that the current low compressibility of the pad results in high pressure variation in the event of the disc being displaced towards or away from it.

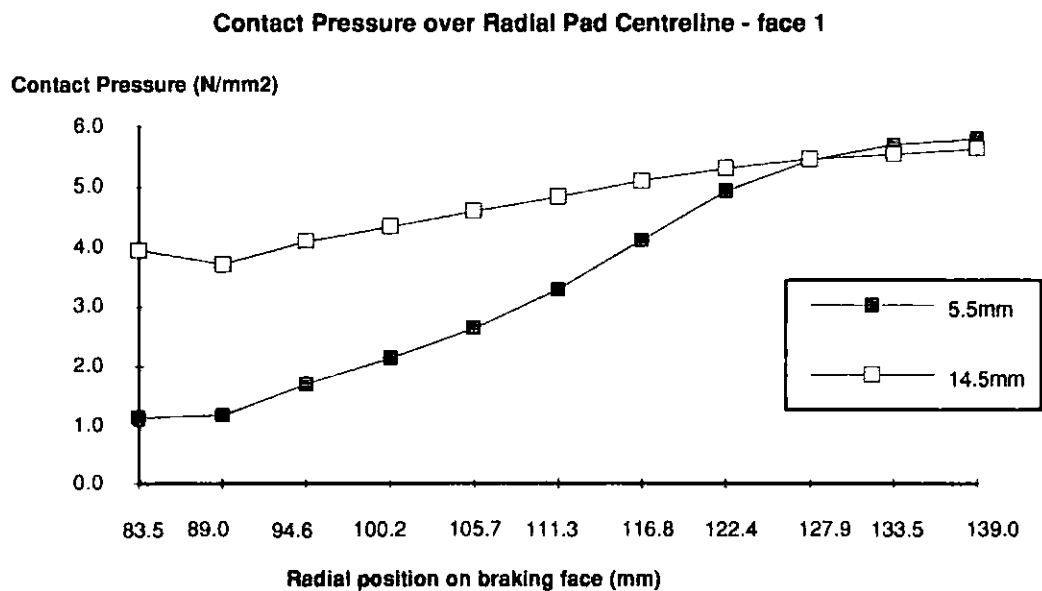


Fig 30. Contact pressure for differing backplate thicknesses - face #1.

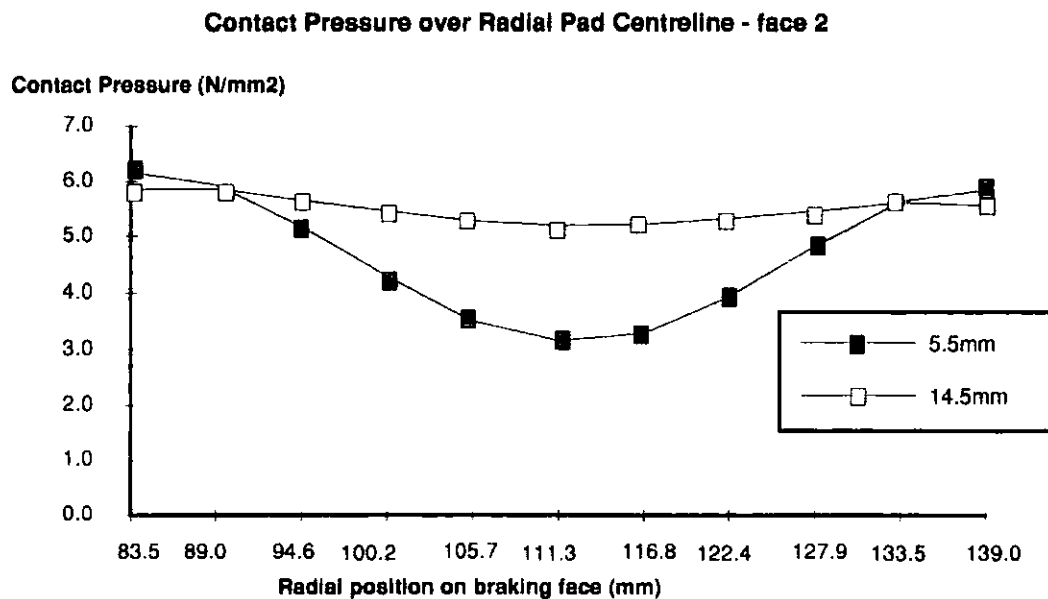


Fig 31. Contact pressure for differing backplate thicknesses - face #2.

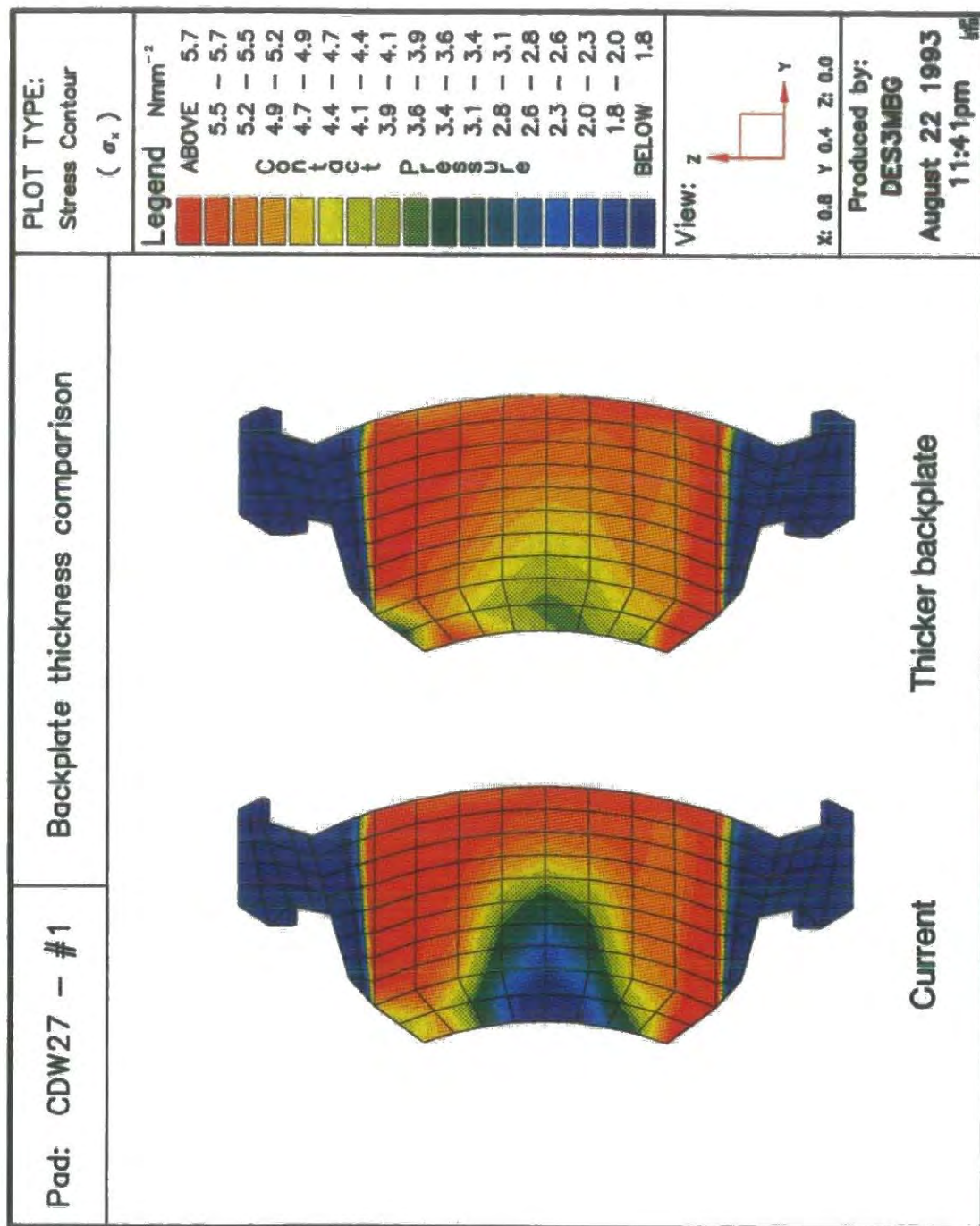


Fig 32. Pad/disc contact pressure distributions - backplate thickness.

5.9 Variable Rib Width

5.9.1 Aims and Procedure

From the original results showing the radial distribution of heat input into the braking faces (Fig 15) it can be seen that there is a marked and progressive increase toward the outside edge of the disc. This is caused by caliper/pad design as already discussed, and the increasing disc speed on an outward radial path. In an attempt to compensate for this uneven input it was decided to try and vary the width of the ribs, increasing their thickness towards the outside of the disc. A single model was developed with a rib width of 5mm at the inside edge of the braking face and a width of 10mm at the outside edge. An 'inside-out' disc was used and it was subjected to the same test conditions as all the other simulations.

5.9.2 Results

An approximate decrease of 6% in the average face temperatures was achieved and the temperature of the ribs was lowered. In addition a 10% decrease of coning angle was observed and a reduction in the spread of the faces by 30%. Radial expansion decreased similarly.

Quantity	Braking Face	Rib Variation	'Inside-out'	Difference (%)
Max. Face Temperature	1	781.3	849.2	8.0
	2	704.6	760.3	7.3
Mean Face Temperature	1	663.2	706.5	6.1
	2	597.2	632.6	5.6
Shoulder Temperature	N/A	118.8	119.3	0.4

Table 18. Temperature summary - rib width variation.

Quantity	Braking Face/Direction	Rib Variation	'Inside-out'	Difference (%)
Coning Angle (deg.)	1	0.43	0.48	9.8
	2	0.47	0.53	11.7
Edge Deflection (mm)	1	0.14	0.17	17.1
	2	0.35	0.40	12.8
Radial Expansion (mm)	1	1.13	1.21	6.7
	2	0.92	0.98	6.0
Shoulder Displacement (mm)	X	0.04	0.04	10.0
	Y	0.47	0.50	5.4

Table 19. Displacement summary - rib width variation.

5.9.3 Interpretation

Increased rib width helped to absorb heat from the braking faces more effectively. The faces were 'cooled' where it was needed most without the corresponding penalty of disc weight increase caused by a general thickening of the rib. However some weight increase is inevitable. The decrease in face spread is significant and is due to the decrease of the thermal gradient along a radial path over the braking face. This design modification is attractive in its reduction of heat input variation caused by poor pad/caliper design.

5.10 Extended Rib

5.10.1 Aims and Procedure

In contrast to the approach taken so far, of decreasing the restraining of the disc in order to reduce coning, it was decided to try an idea concerned with restraining it more so that the physical displacement required to cause coning was rendered impossible. The method of achieving this was to extend the rib towards the hub of the disc until it joined the disc wall. In this way the shoulder right angle would not be allowed to open and an improved thermal path from the faces to the cool hub could be created. This design consisted of a modification to the 'inside-out' design and the usual loads were applied.

5.10.2 Results

Only marginal differences were apparent in the disc temperatures, even though a very different thermal path is in effect. The coning results showed a slight reduction in absolute value but a reversal of the direction in which they acted.

Quantity	Braking Face/Direction	Extended Rib	'Inside-out'	Difference (%)
Coning Angle (deg.)	1	-0.46	0.48	4.8
	2	-0.31	0.53	40.9
Edge Deflection (mm)	1	-0.64	0.17	-277.1
	2	-0.41	0.40	-2.7
Radial Expansion (mm)	1	0.93	1.21	23.4
	2	1.10	0.98	-12.7
Shoulder	X	0.00	0.04	90.0
Displacement (mm)	Y	0.54	0.50	-7.0

Table 20. Displacement summary - extended rib model.

5.10.3 Interpretation

From the coning results it was deduced that the disc section in the region of the shoulder had simply been made too rigid, the angle unable to open and compensate for the wall deflection. A revision was made to the model, decreasing the extent of the rib/wall connection, which effectively reduced the rigidity of the section in this region and the simulation was rerun. Figs 33-35 show temperature and stress distributions and the displaced shape plot. Again there was no significant change in temperatures but the disc coning was reduced to virtually zero (face #1 - 0.0° throughout the stop, face #2 at 0.13°). However the stresses developed in the shoulder were extremely high. This is because, as described earlier, the magnitude of the expansion force is so great that any attempt to restrain it will result in far higher stresses than those incurred by simply allowing the disc to expand naturally. These high stresses are very undesirable, local plastic deformation almost certainly being encountered and the likely source of potential disc cracking. It was therefore decided that although a massive reduction in coning had been achieved, it was less desirable than the previous method, using unequal face thicknesses, which developed far lower stresses. In

addition the likely unfavourable cooling properties of the 'inside-out' disc also influenced this decision¹.

Quantity	Braking Face/Direction	Extended Rib	'Inside-out'	Difference (%)
Coning Angle (deg.)	1	0.00	0.48	99.8
	2	0.13	0.53	75.7
Edge Deflection (mm)	1	-0.14	0.17	20.0
	2	0.09	0.40	76.8
Radial Expansion (mm)	1	1.05	1.21	13.4
	2	0.99	0.98	-1.4
Shoulder Displacement (mm)	X	-0.02	0.04	62.5
	Y	0.51	0.50	-1.2

Table 21. Displacement summary - revised extended rib.

It was also felt that the zero coning value was the result of good fortune rather than good design. There was no variable to be changed and it was perhaps lucky that this fixed design came out with some of the desired properties. This does not mean that the same benefits would be consistently, if at all, evident when an actual disc is mounted on a vehicle.

¹ It was interesting to note, some time after this testing had taken place and the conclusions drawn, that the Lotus Elan and certain high performance variants of the Vauxhall Cavalier both employ ventilated brake discs on their front axles of a very similar design to those described in this Section.

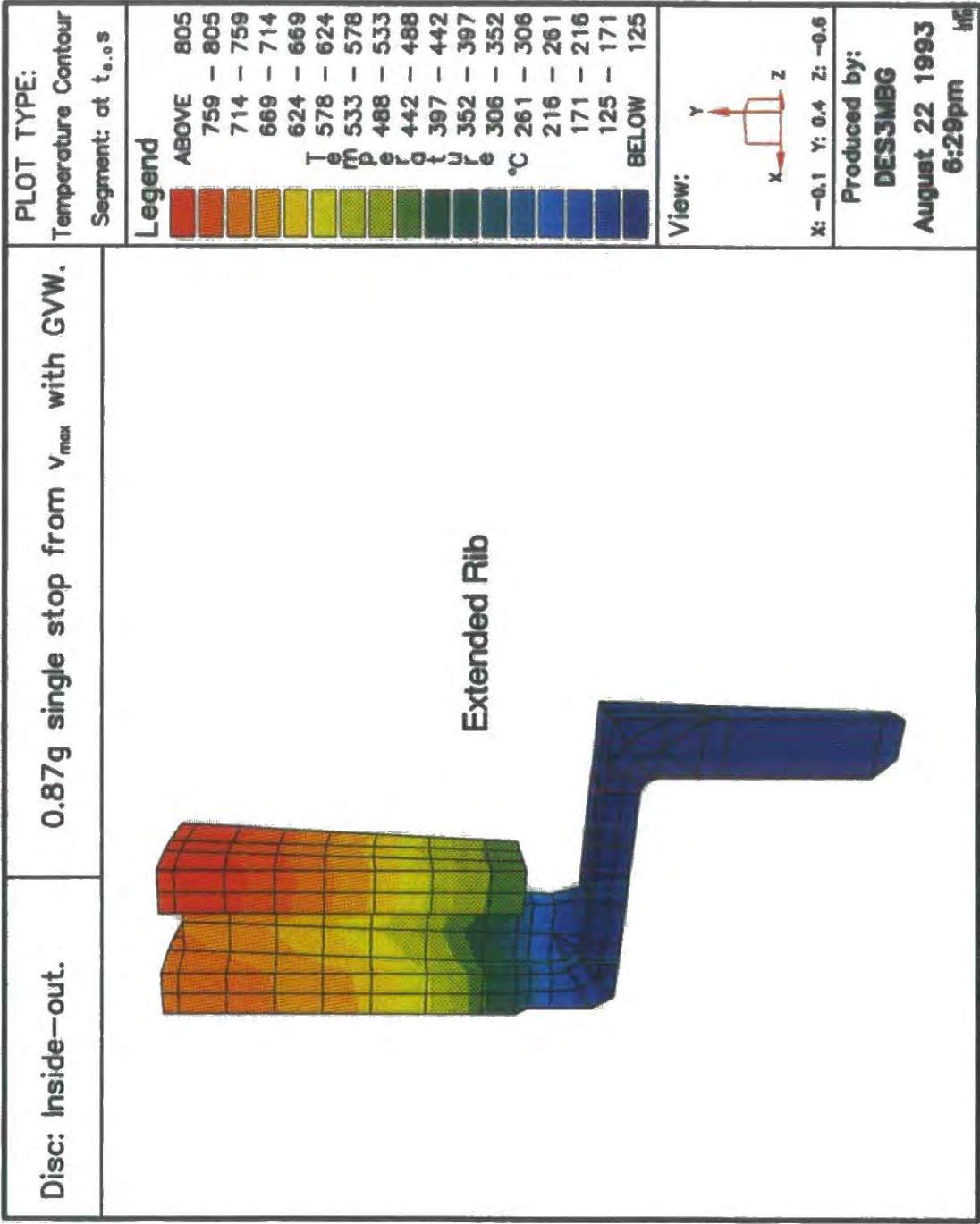


Fig 33. Temperature distribution - revised extended rib.

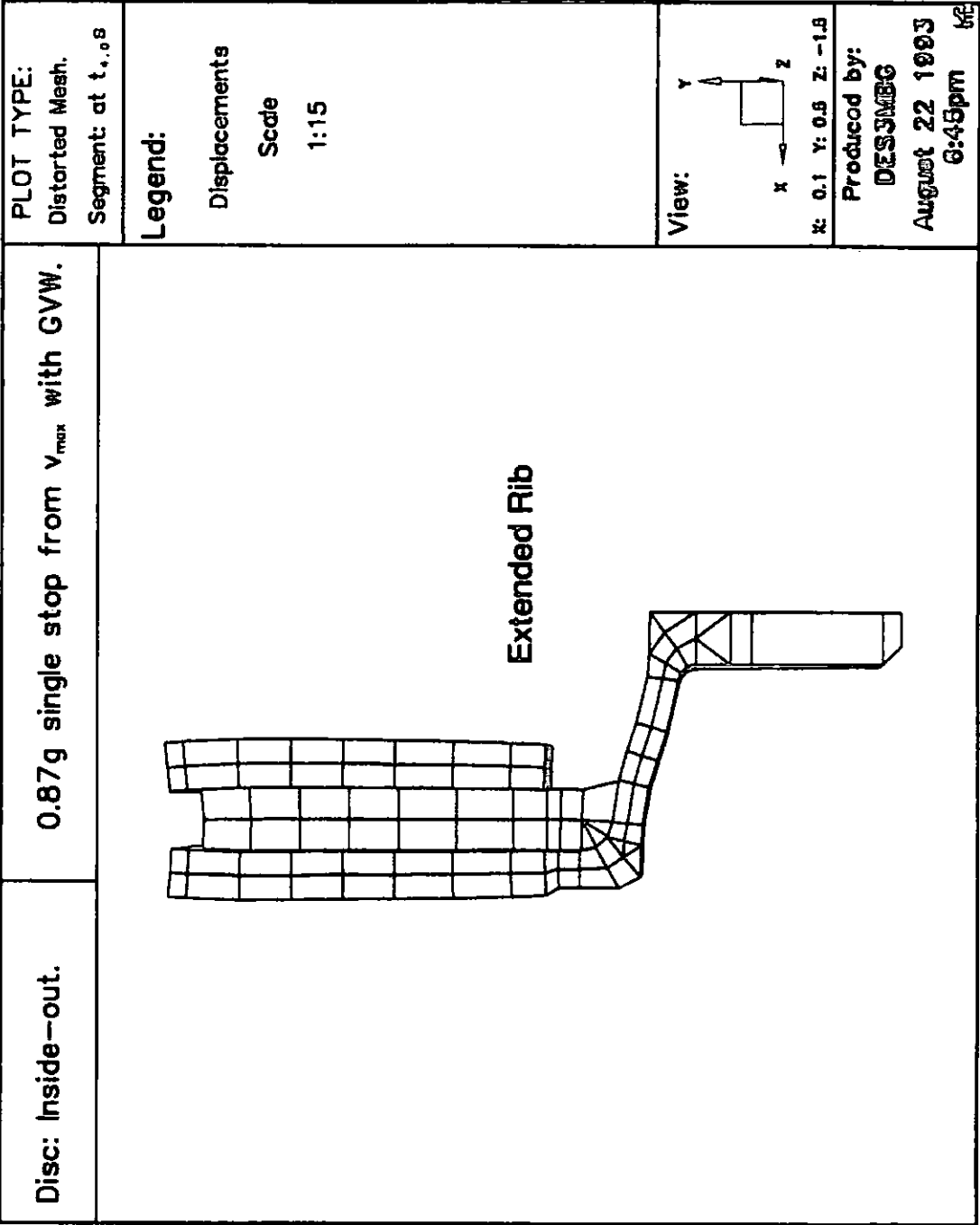


Fig 34. Displaced shape - revised extended rib.

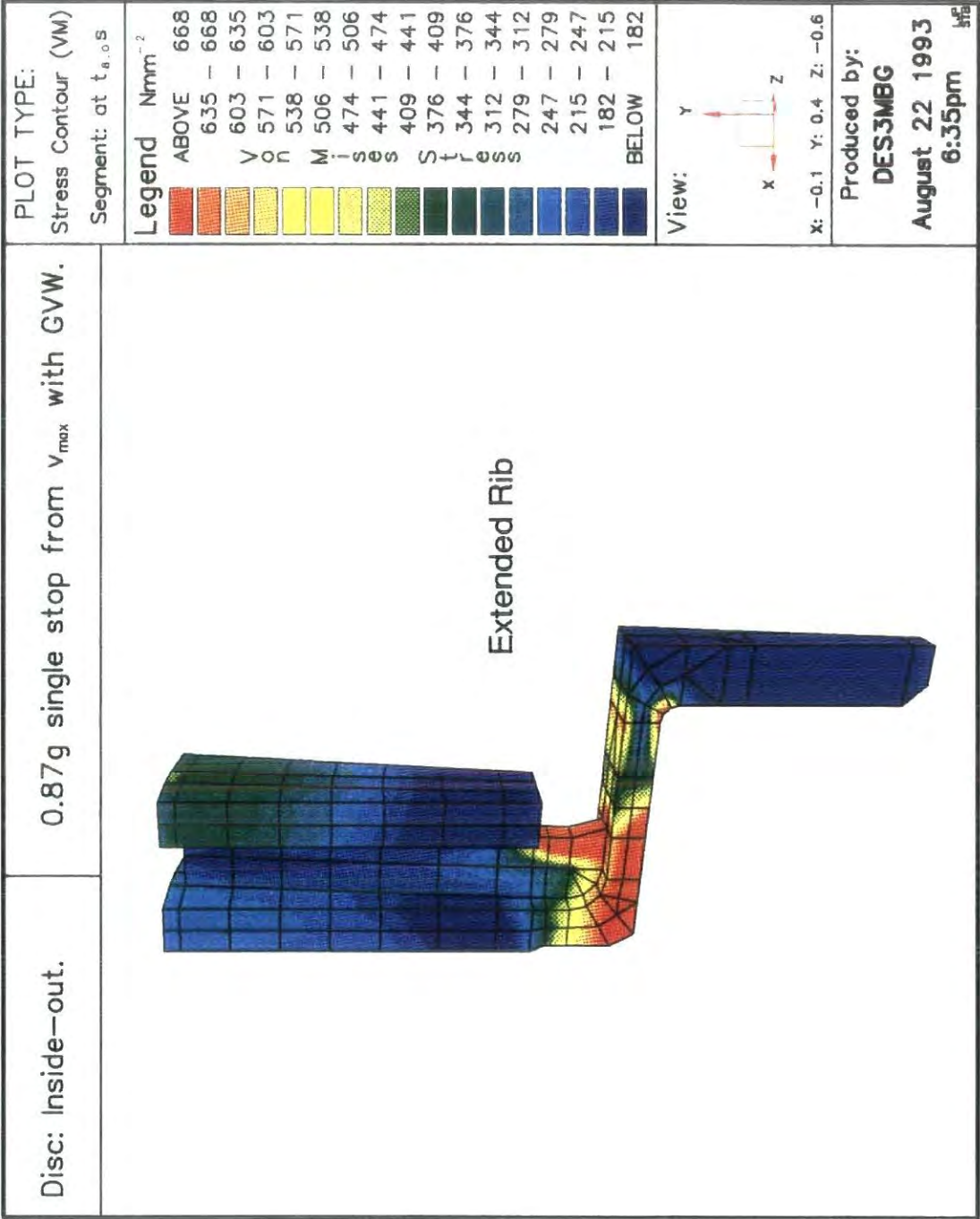


Fig 35. Stress distribution - revised extended rib.

5.11 Curved Shoulder

5.11.1 Aims and Procedure

Returning to the original philosophy of reducing the restraining forces to reduce coning it was decided to produce another shoulder design in complete contrast to the one above. It was attempted to make the shoulder more flexible and more willing to open out by removing material from the external corner of the shoulder, lightening the section. It was hoped that this 'weakening' of the section would result, not in a reduced stiffness of the disc wall but an increased inclination for the shoulder angle to increase, lowering the restraining force able to be imparted by it. It was decided to try it using a conventional disc, the attractive cooling properties being preferred.

5.11.2 Results

The temperatures were found to be practically identical. The reduction in coning was approximately 5-6% with a 4% increase in radial expansion as expected. Most importantly the stresses in the shoulder section were significantly reduced.

5.11.3 Interpretation

Although the distortion reduction is not great, the most important thing is the large reduction of stresses. An apparently 'weaker' section gives *less* distortion and a big reduction in the stress concentration. It was decided to take the idea further and combine it with a deeper disc section. The length of the wall was increased by 17mm and the simulation rerun. This time a coning reduction of between 16% & 19% was observed in combination with an 8% increase in radial expansion (this value was still sufficiently small to have no bearing on the function of the disc). Even more improvement in the stress concentration at the shoulder was evident (see Fig 36) but the most significant effect is because the length of the disc wall has been increased, the stress concentration at the corner of the hub is almost removed (c.f. Fig 17). This design is pleasing as excellent results are achieved using a conventional disc (therefore good cooling performance) and the large reductions in stress concentrations and coning are not accompanied by any increase in disc mass.

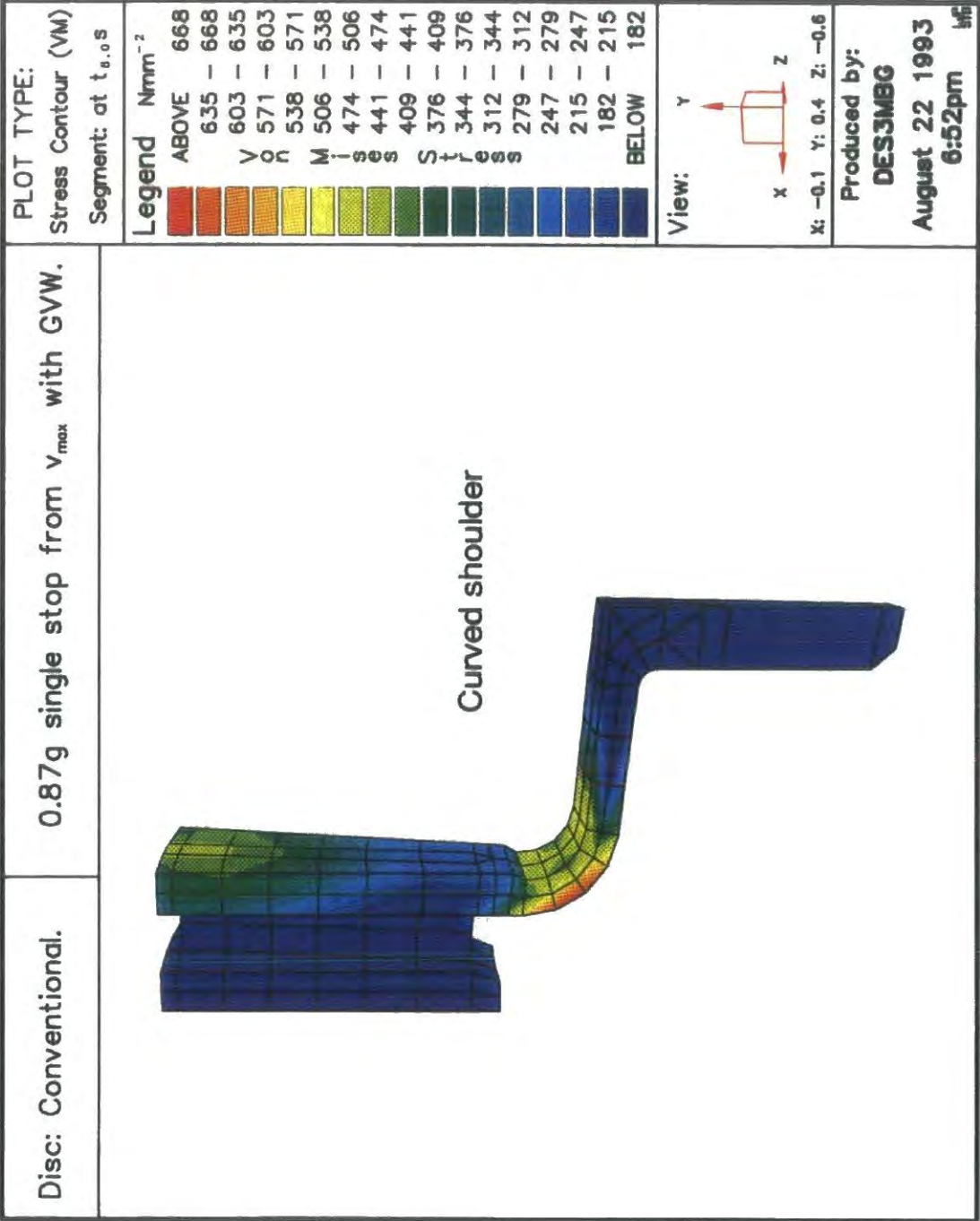


Fig 36. Stress distribution - curved shoulder model.



It was then felt that the next step should be the necessary incorporation of some cooling trials (by convection and/or conduction) into the simulations. Although the limitations of the available cooling equations were discussed earlier, it was decided to try and use them on the grounds that if the results were in the right order of magnitude then it would not be necessary to have very accurate cooling values for comparison of models. The variation of convective cooling conditions surrounding a disc in service is such that no definitive correct values could exist anyway. A common measure of cooling was required and this was supplied by the equations in Appendix V.

5.12 Disc Cooldown

5.12.1 Aims and Procedure

It was first decided to compare disc cooling by conduction only with cooling achieved by both conduction and forced convection of the spinning disc. Both disc types were incorporated into the simulation and so four models were developed. The disc models were heated using the usual single stop conditions and then a ten minute cooldown was simulated, the discs incorporating forced convection consistent with a rotation of at 34.5Hz. (v_{max}) in air at ambient temperature.

5.12.2 Results

The average face temperatures are presented in the table below. In both convection and no convection cases the conventional disc design cooled to a temperature 17% lower than the 'inside-out' design.

Time (min)	'Inside-out' - air cooled	Conventional - air cooled	'Inside-out' - no cooling	Conventional - no cooling
2	490	482	590	580
4	375	353	540	508
6	289	259	492	444
8	223	192	450	389
10	174	144	411	342

Table 22. Cooldown test results.

5.12.3 Interpretation

Air passage cooling was not included and it is likely that this would have been in the favour of the conventional disc because of the more simple route of cooling air through the disc. However even before this was included the conventional disc still performed much better. In addition the difference was the same in both cooling cases and so must have been independent of forced convection (the conditions of which were effectively the same in both cases anyway). This 17% improvement is due to the shorter conductive path of the conventional disc between its faces and the disc hub. Ford tests have shown a 25-30% improvement in cooling of conventional over 'inside-out' discs and it is possible that the 8-13% difference is due to the increased effectiveness of the conventional disc to lose heat through its air passages compared to the 'inside-out' disc.

These results were also extrapolated to provide a time to cooldown to 100°C. In this case the convection run, the conventional disc cooled 5% faster whilst for the no convection case the conventional disc cooled 19% faster (the similar Ford test for Cosworth Scorpio discs from 250kph results in a 15% improvement of the conventional disc over the 'inside-out' disc).

5.13 Drag Test

5.13.1 Aims and Procedure

This test was devised to obtain a measure of the disc behaviour during a multi-stop braking schedule, where a condition approaching steady state conduction and convection is achieved. The test devised is known as a drag test and can be performed on vehicles by driving with the brakes partially applied. It was decided to apply a steady flux to the braking faces to maintain the braking faces at between 350°C and 400°C. It is simplistic but is easy to model and is likely to produce results that are similar to those obtained during a course of many brake applications, for example during the descent of a mountain pass. Both disc designs were tested.

5.13.2 Results

The temperatures on the conventional disc were 12.5% lower than those on the 'inside-out' disc. However the coning on the 'inside-out' disc was far better, a value of 0.10 being obtained compared to -0.33° for the other disc.

Quantity	Braking Face	'Inside-out'	Conventional	Difference (%)
Max. Face Temperature	1	403.1	356.1	11.7
	2	398.5	356.2	10.6
Mean Face Temperature	1	391.9	323.1	17.6
	2	373.3	345.3	7.5
Shoulder Temperature	N/A	262.4	193.0	26.4

Table 23. Drag test temperature summary.

Quantity	Braking Face/Direction	'Inside-out'	Conventional	Difference (%)
Coning Angle (deg.)	1	0.10	-0.36	-260.6
	2	0.10	-0.32	-234.4
Edge Deflection (mm)	1	-0.01	-0.49	-3392.9
	2	0.11	-0.38	-257.0
Radial Expansion (mm)	1	0.67	0.50	25.9
	2	0.63	0.63	0.2
Shoulder Displacement (mm)	X	0.09	0.06	30.7
	Y	0.32	0.25	21.3

Table 24. Drag test displacement summary.

5.13.3 Interpretation

The improved cooling was due to the better thermal path of the conventional disc, but like the standard single stop test, the 'inside-out' disc is better designed to cope with coning. The values are roughly a third of those obtained on a single stop, the radial expansion of the disc being approximately half that observed in a single stop. These values were useful for later comparison with the proposed 'improved' design.

The next series of simulations were conducted with full disc models and usually concerned quantities that varied with the angular position on the disc face. The models took correspondingly longer to run.

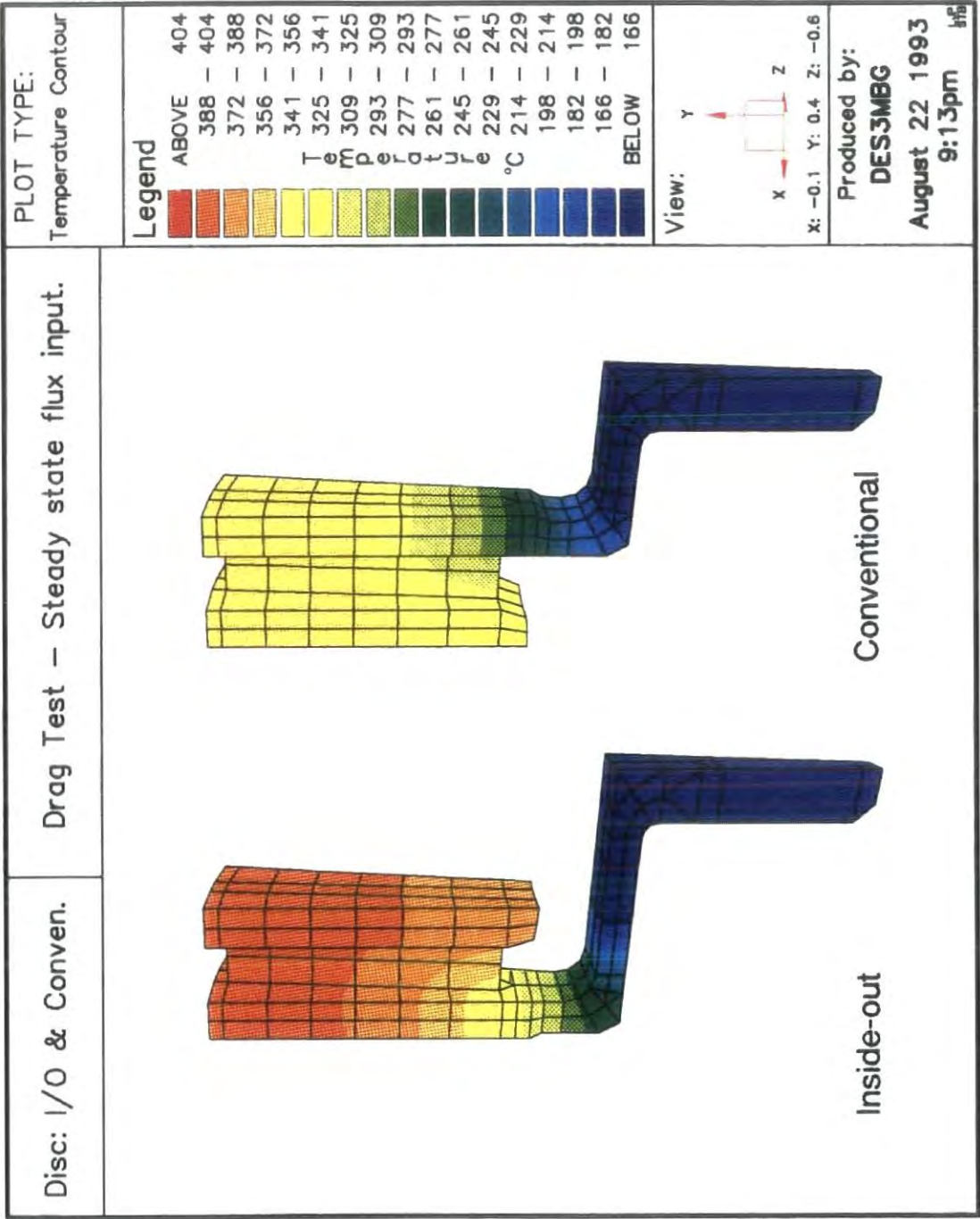


Fig 37. Temperature distribution - drag test.

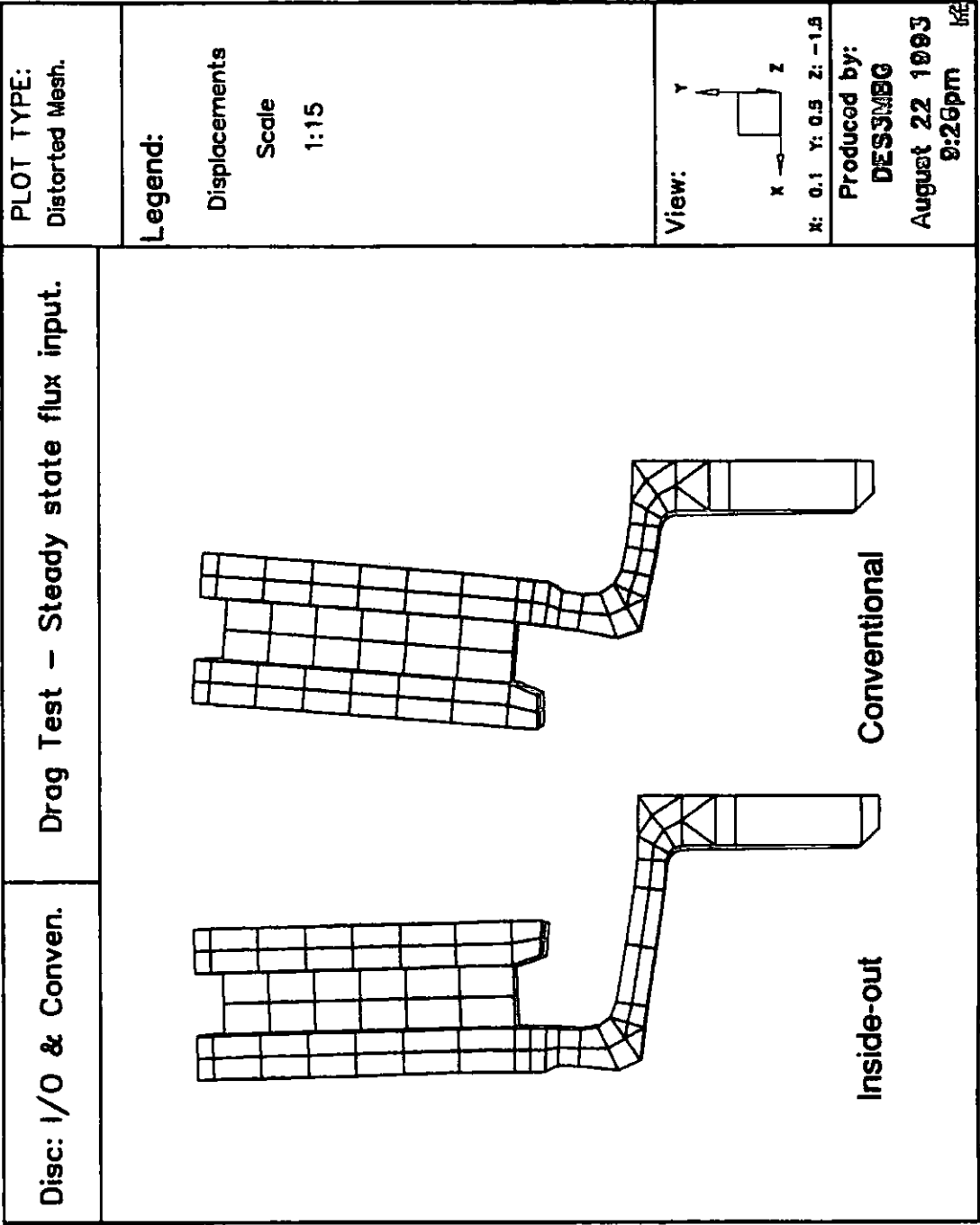


Fig 38. Displaced shape - drag test.

5.14 Full Disc

5.14.1 Aims and Procedure

Initially two full discs were run, one 'inside-out' and one conventional, to provide reference values for further testing. They were subjected to the same heat input, from the single stop test, but were also subjected to pad loads. These pad loads comprised both frictional and axial loads and were applied over the correct pad area on the disc. The disc was modelled with linear elements (instead of quadratic) for reasons explained earlier.

5.14.2 Results

The temperatures observed were slightly higher than those obtained for the disc segment under the same conditions. In addition to the coning, an axial displacement wave was observed in both cases around the braking face and this is shown graphically below. In the 'inside-out' disc the amplitude was approximately 0.07mm with the conventional disc showing 0.1mm.

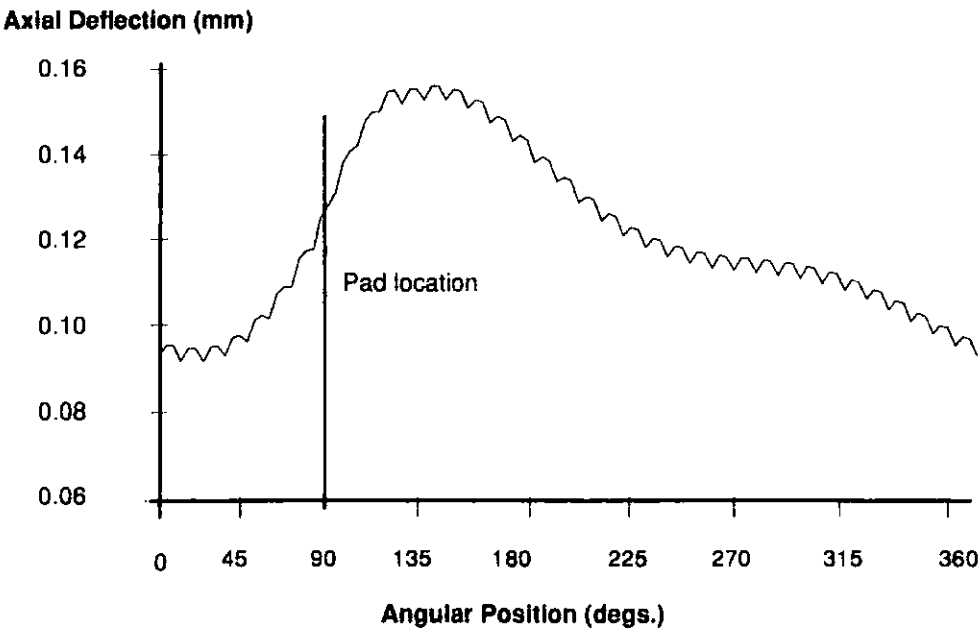


Fig 39. Variation of axial deflection with angular position - i/o.

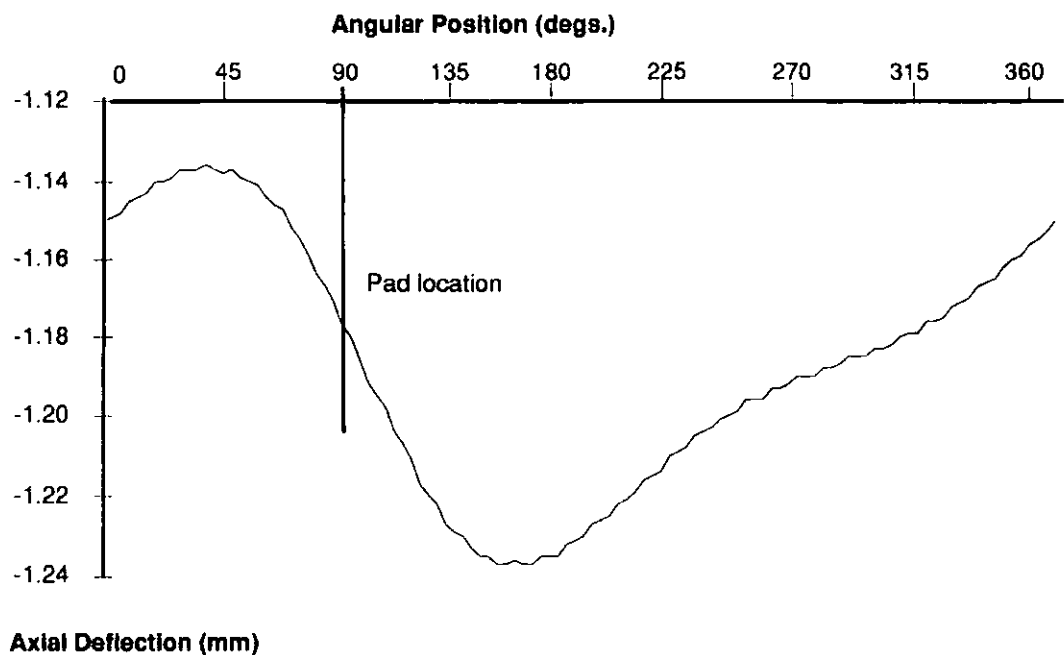


Fig 40. Variation of axial deflection with angular position - conv.

5.14.3 Interpretation

The slight difference in temperatures was probably due to the difference in modelling elements. As the only difference to the segment case was the addition of pad loads it is apparent that the waving is caused by the mechanical braking loads being placed on the disc by the pad. The inside out disc appears to show slightly increased ability to withstand these loads but in order to understand why it was decided that this effect and its causes would be considered more closely in subsequent simulations. The existence of a wave is consistent with Inoue's measurements of a residual plastic deformation on discs subjected to heavy braking. However Inoue found a double wave existing around the disc whereas the distortion experienced in this case was, for both discs, a single wave. The apparent rippling superposed over the main waveform is localised deformation of the rotor cheek between ribs. This, although a type of distortion, has very little effect on the braking performance because of its minimal amplitude. It is possible that it may contribute to squeal or brake noise, its 37 peaks per revolution causing high frequency vibrations to be set up in the brake components.

5.15 Heat Input Variation

5.15.1 Aims and Procedure

An amount of runout on the disc, caused by inevitable inexact mounting, would cause a variation of pad pressure on the disc surface as it rotates and this variation produces a corresponding heat variation. In this case it was decided to try and examine the effect of a varying heat input around the disc braking face. The precise conditions of the pad pressure variation are unknown and difficult to quantify and so very simple values were chosen, with no attempt to model a specific case but to demonstrate the effect of heat variation. The standard heat inputs were manipulated to produce a variation of $\pm 25\%$ at a frequency of 2Hz around the disc. This heat input variation was largely arbitrary but it was known that because of pad material compressibility, this was not an unreasonable value as the known runout of 0.07mm was much larger than the displacement known to cause a 25% increase in pad pressure using current pad material compressibility (Appendix I).

5.15.2 Results

Fig 41 shows the resulting temperature variation around face #2 of the disc. The section also shown in the figure illustrates how a hot region on one face of the disc was located opposite a cool region on the other face. Both discs exhibit a temperature variation of about $\pm 16\%$ around the face. A 'wave' of the same phase and frequency as the heat input was observed as expected. The resulting displacement wave around the disc is shown in Fig 42. The 'inside-out' disc had a wave amplitude of 0.6mm and the conventional was 0.5mm and it can be seen that although the wave mean values and the amplitudes are different they are entirely in phase with one another and the inputs.

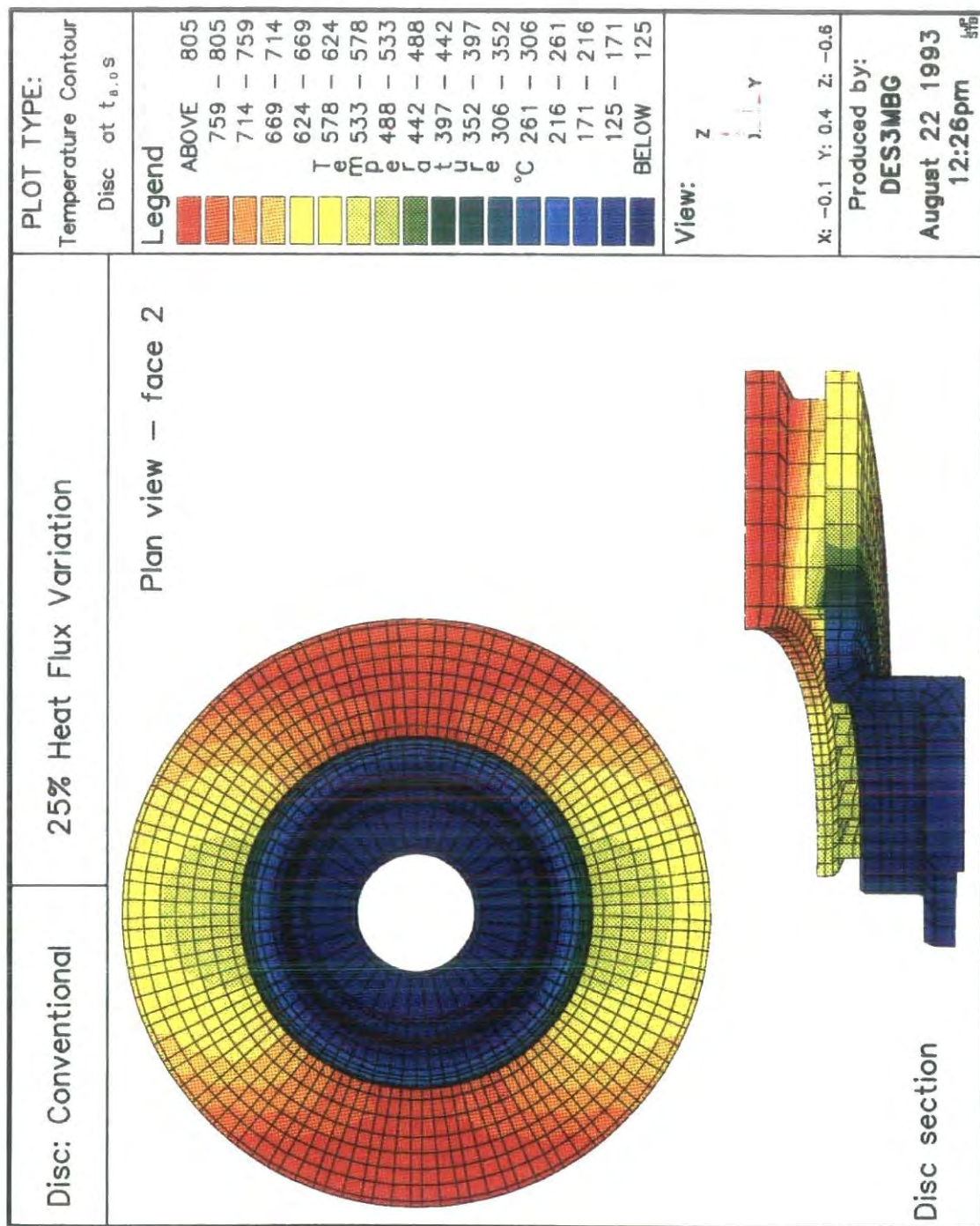


Fig 41. Temperature distribution - varying heat input.

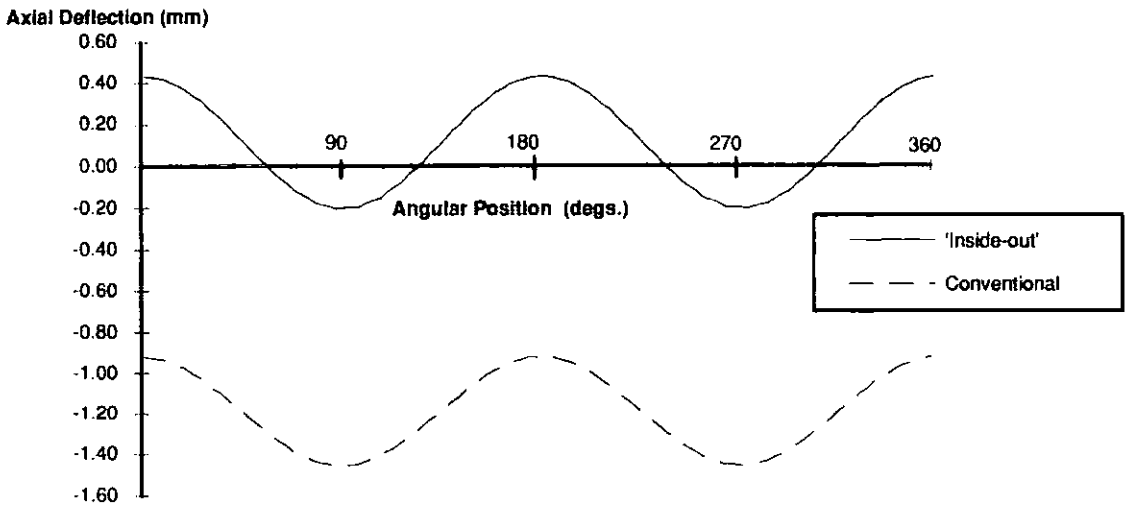


Fig 42. Axial deflection of braking face with varying heat input.

5.15.3 Interpretation

The distortion wave is as expected. The opposing heat variation (a maximum on one side of the disc corresponding to a minimum on the other because of the nature of runout) causes a variation of the angle of coning at any one point. It is precisely this variation of coning with angular displacement which defines a wave deformation of a brake disc. The amplitudes experienced are of a much larger order than runout or plastic distortion would be which tends to suggest that the effect could be unstable or self-increasing. Therefore a small value of runout or residual deformation from a particularly heavy brake application could result in this heat variation being produced resulting in a wave which in turn causes an increase in the variation of pad pressure and heat input as the disc rotates. A self sustaining growth of deformation would then occur until some stable point was reached where either heat loss or mean pad pressure became stable and governing. As it could be seen that growth of wave deformation might be self-exciting once initiated it was decided to examine further the causes of the initial 'seed' of this effect. This could either come from runout or from a plastic deformation wave caused by excessive mechanical/thermal loads as observed by Inoue. As runout has already been examined in effect it was decided to look further into disc distortion under plain mechanical loads with a view to what design factors affect its presence and the improvements in design that can be made to reduce it.

5.16 Mechanical Load Deformation

5.16.1 Aims and Procedure

In order to produce a set of reference values two models were created of full discs but only being subjected to mechanical pad loads, again one 'inside-out' and one conventional. In order to further isolate causal factors a further two similar models were generated. These were only subjected to frictional loads from the pads, the axial 'squeezing' loads being removed. The four discs were subjected to the standard single stop braking loads.

5.16.2 Results

The graphs in Figs 43 & 44 show the results observed. The waves take the form of a single wave around the braking face, but with a shoulder occurring approximately opposite the pad location. The waves have maximum gradients at the pad location, meaning that at this position the disc is not running true and would appear to be running through the caliper at an angle. This is illustrated in Fig 45 where the edge of the disc is viewed at the caliper position. The waveforms ('inside-out' compared to conventional) also have similar shapes but are reflected in the 'y' axis about the pad location point. This can be seen by comparison of Figs 43 & 44. The mean of the graphs (and also the deflection at the pad location) of those models *with* axial loads is displaced from zero in a positive direction, indicating positive coning in both disc types. Both models *without* axial loads show no such mean displacement and the deflection at the pad location is zero. The amplitude of the wave is greater in both cases for the conventional disc although where applicable its displacement from zero is less.

A radial displacement was also noted, a waveform again being observed around the edge of the disc. This is also a single wave and is displaced in a negative direction. The displacement values are provided in the tables.

Axial Deflection at Outer Edge vs. Angular Position

Axial Deflection (mm*1000)

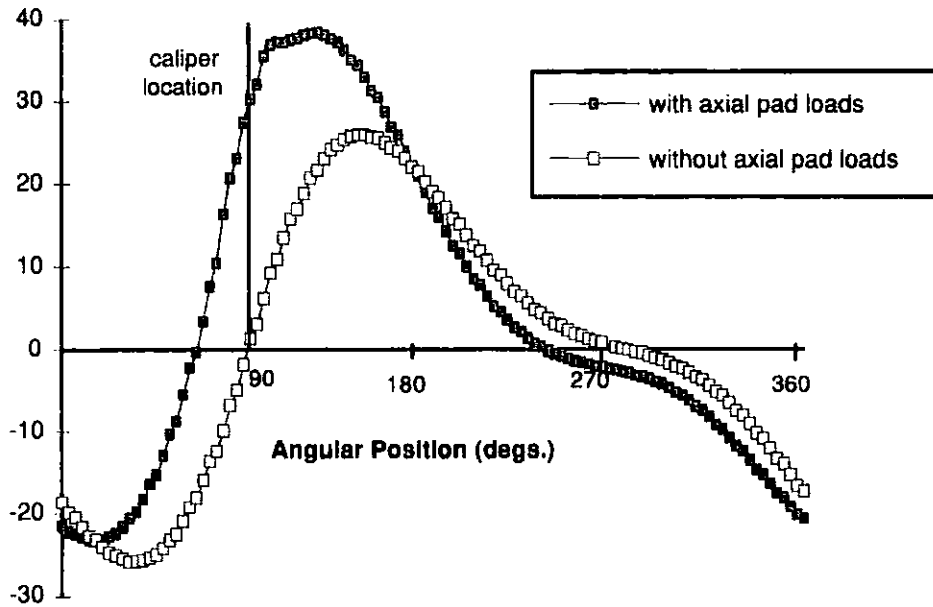


Fig 43. Axial deflection due to mechanical loads only - i/o.

Axial Deflection at Outside Edge vs. Angular Position

Axial Deflection (mm*1000)

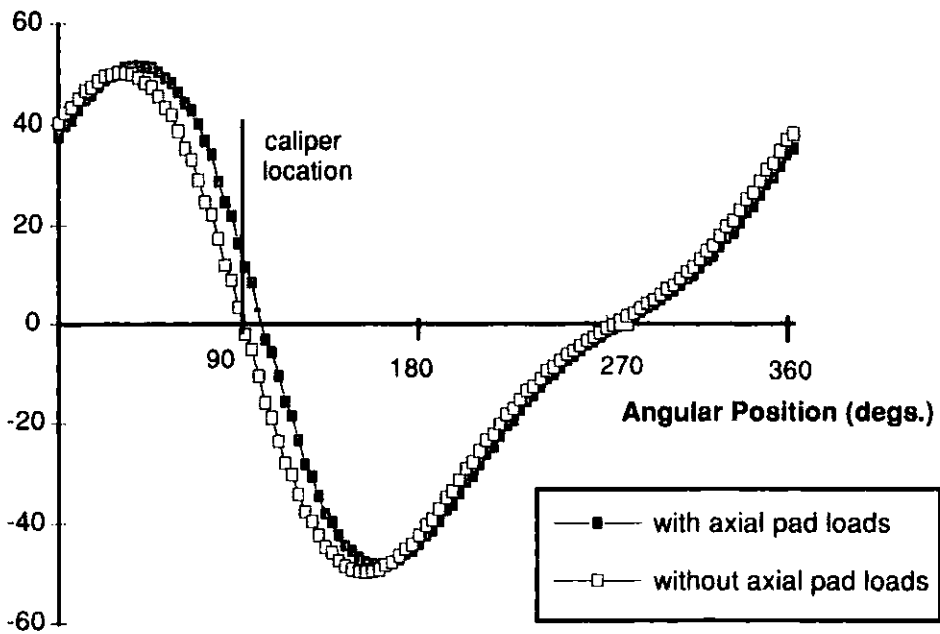


Fig 44. Axial deflection due to mechanical loads only - conv.

5.16.3 Interpretation

Firstly, the absence of axial loads removes the offset of the graph and so, superposition applying as found, the slight positive coning (particularly at the pad location) is due to these squeezing loads. The total loads are equal but it has been well discussed already that the pressure distributions are different and it is apparent that the centre of pressure of pad #1 is much closer to the outer edge of the disc than pad #2. The magnitude of these pad loads is such that it is suggested that the couple produced by the unequal positioning of the pressure centres causes a resulting deflection. In both disc designs the resulting displacement is positive, consistent with this hypothesis. The removal of these loads, results in the removal of the displacement.

Secondly, comparing conventional graphs to 'inside-out' graphs in all cases the curve could be seen to 'pivot' about the pad location (angular position of 90°). The implication is that a similar reaction is occurring but that there is a different imbalance in load application. Something has to make the braking face 'buckle' one way or the other and this direction appears to be consistent throughout the tests for a given design, and so it was felt that the disc design influenced this.

Thirdly, as already mentioned, the gradient is such as to imply that the friction pad loads themselves are unbalanced and exerting a twisting effect (of opposite sense in the two types of disc).

The following explanation was developed from the results of this test. The imbalance is dependent on disc design and the only difference in design was the orientation of the faces relative to the shoulder ('inside-out' or conventional). If the disc is viewed from the outside at the pad location point even frictional forces are being applied to the two faces. However, considering a segment for example, one of the faces is restrained and the other is free and so a twisting effect is observed where the free face displaces further in the direction of the force. In this way a wave is initiated as the disc passes through the caliper, its initial direction governed by whether it is the upper or lower face (when viewed as in Fig 45) that is connected to the wall. Fig 46 illustrates this mechanism where an imbalance is created by the natural fulcrum of rotation always being offset towards one of the two *even* pad frictional forces. If a steady deformation is to result as the disc is rotating then the disc must enter the caliper at approximately the same angle at which it leaves it and so it is necessary for

the disc to assume a waveform deformation as it first tries to recover its equilibrium and then is forced into approaching the caliper at an angle. Like coning, the magnitude and sense of this effect is governed by the lack of symmetry inherent in the section of the ventilated disc and hence its direction is reversed when the disc type is reversed. In addition this imbalance is likely to exaggerate any buckling tendency that the disc faces may possess as they enter the caliper, there being large compressive forces in a circumferential direction in this region.

However, it can be said that as this wave is not the major deformation itself but is the cause of it, it is desirable to reduce it. If, as previously said, the unstable self-excitation of the heat input variation waving requires only a small amount of this initial seed then there is an argument stating that it is immaterial how much it is decreased because sooner or later it will build up if any initial waving exists at all. It is not possible to rid a disc of this kind of buckling entirely and a perfect disc mounting on the vehicle (no runout) cannot reasonably be achieved in production. This creates an amount of inevitability that thermal waving will build up; greater reductions in runout or plastic deformation only resulting in the process taking longer.

It is possible that the greater waving amplitude in conventional discs is caused by the lower resistance to twisting of the shorter disc wall length. However the greater axial displacement of the 'inside-out' disc at the pad location caused by uneven pad loads would suggest that the wall length is a weakness in this case, the greater flexibility, desirable for coning reduction, proves to be undesirable in this instance.

It was decided to conduct a further series of tests, examining the effects of some different dimensions on the extent to which this mechanical waving occurs. It must be borne in mind that the results given above are for heavy brake application (although possibly not the maximum achievable) and are elastic.

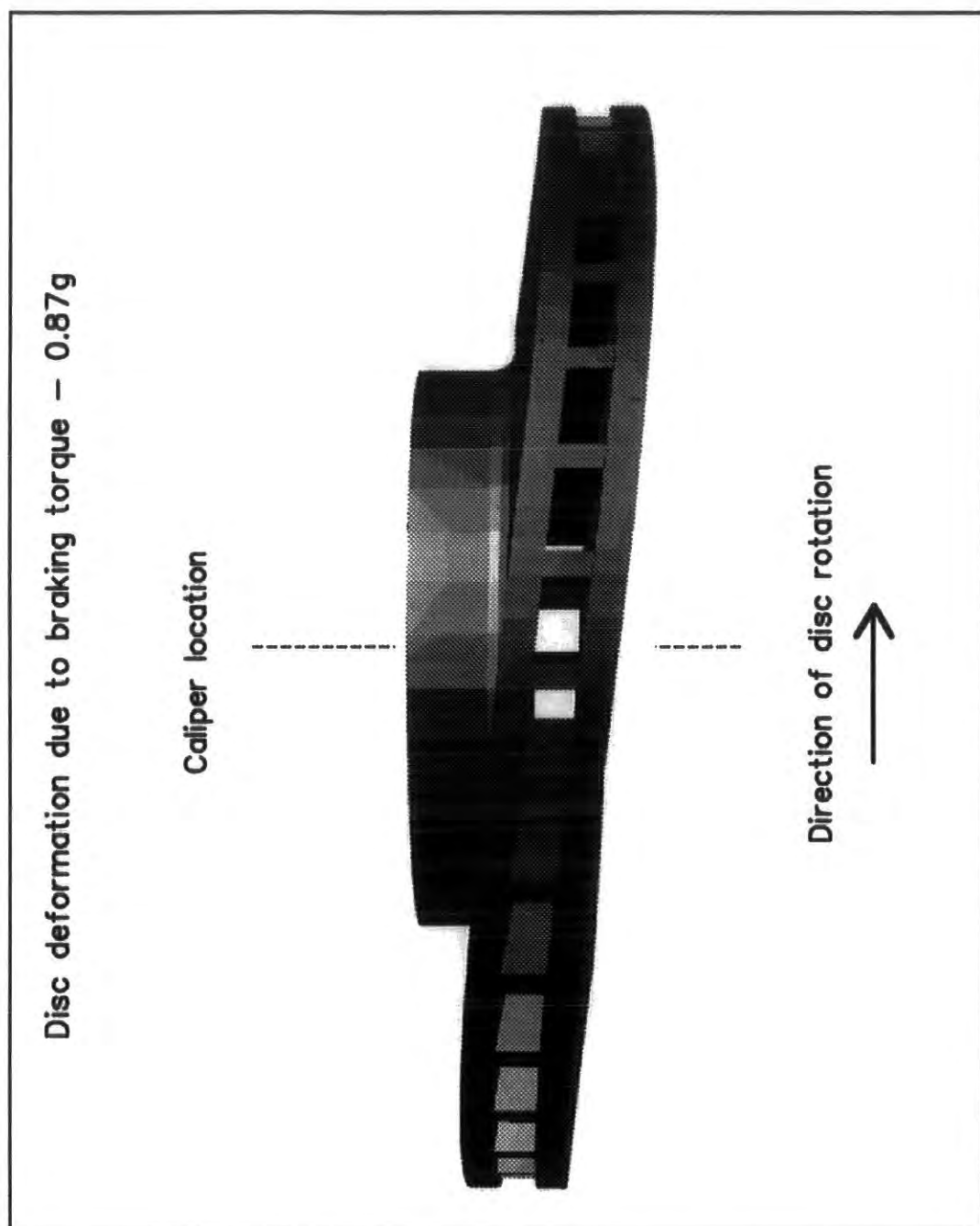


Fig 45. Twisting effect causing wave distortion of braking faces.

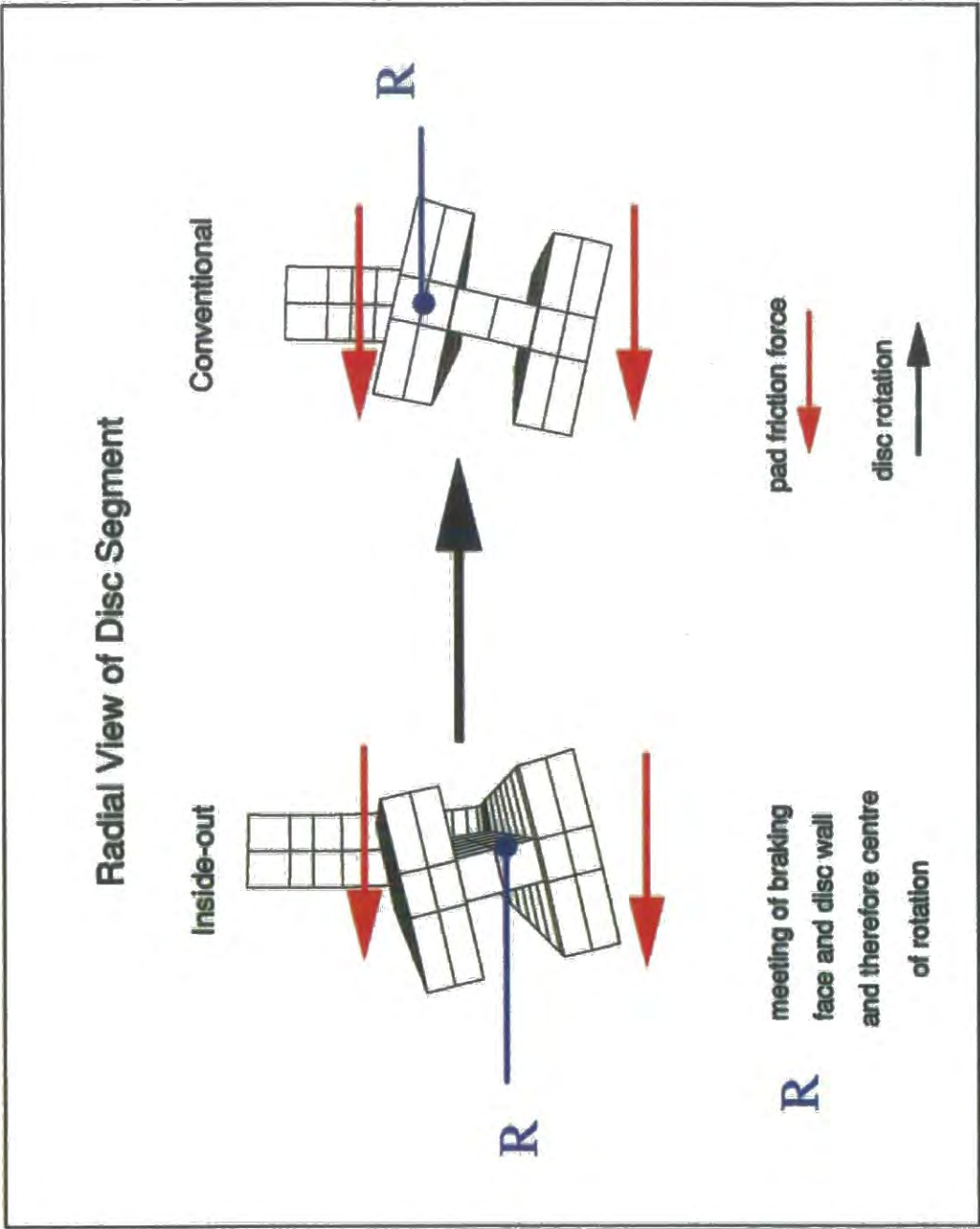


Fig 46. Mechanism of 'twist' caused by friction loads.

5.17 Reduction of Mechanical Waving

5.17.1 Aims and Procedure

Two further pairs of disc models were generated. They were chosen to demonstrate the effect of potential improvements to disc design and to test the hypothesis of mechanical deformation due to braking loads. The first pair of models (one conventional and one 'inside-out') had the unrestrained face thickness expanded to 10mm, the measure discussed in Sections 5.3 & 5.4 to reduce coning. The second pair of discs were modelled with the air gap between the faces changed from 10mm to 15mm. The discs were again subjected to mechanical pad loads only.

5.17.2 Results

The table below shows the behaviour differences obtained.

Model	Type	Waving Amplitude (mm)	Face Displacement (mm)
Unaltered	i/o	61	30
	conv.	100	15
Uneven Face Thickness 7mm and 10mm	i/o	55	22
	conv.	95	8
Large Air Gap 15mm	i/o	70	25
	conv.	105	10

Table 25. Effects of design changes on mechanical deformation.

5.17.3 Interpretation

The uneven face thickness results will be examined first. A slight reduction in waving amplitude was achieved. Although the increased thickness also increased the twisting couple developed by the frictional forces, it tended to stiffen the face reducing its tendency to buckle. A significant decrease in the coning resulting from the uneven pad loads was also as a result of the stiffness being improved.

In the case of the air gap, although the axial deflection was reduced, there was an increase in waving amplitude because, as with the uneven face thicknesses, the increased separation of the friction forces caused a larger couple to be generated and a greater gradient of the face at the pad location.

Both sets of results were consistent with the hypothesis of mechanical disc waving. It was encouraging to see that the uneven face design, chosen for coning prevention, also provided a positive contribution to the reduction of the potential 'seed' of thermal waving, plastic deformation caused by high pad loads.

The 'inside-out' disc tended to exaggerate the benefits or drawbacks, a lesser benefit or detrimental effect always being experienced by the conventional disc design. The implication that this design is more stable is attractive as its recommendation is more and more assured.

5.18 Ford Experimental Work

Some experimentation was completed using the dynamometer and recently installed thermal imaging equipment at Ford. The test procedure was specified as being identical to that employed within the computer simulations and the disc, pads and caliper used were the same. This work was undertaken shortly after the experimentation detailed earlier in the Section. Appendix VI contains the practical results obtained and some analysis of them. They are referred to in the following Chapter.

CHAPTER 6

RESULTS ANALYSIS

6.1 Disc Distortion Hypothesis

The following is a description of the process and mechanism of disc deformation as observed during this work. It enables the brake designer to understand what conditions are being designed for and to be aware of some of the functions of the disc.

The principal types of deformation are waving and coning, these being the ones that cause audible vibration and reduced braking efficiency. Waving appears to be caused by varying heat input around the disc face and therefore waving can simply be described as a variation of coning angle with angular position on the disc face. Therefore the primary mechanism of distortion is coning.

However it has also been suggested that the cause of varying thermal input is an initial mechanical condition. This is most likely to be a combination of disc mounting runout and a plastic deformation in the form of a wave as shown by the displacement graphs in Section 5. Runout is inevitable in a mass-production environment and it should not be assumed that it could be eradicated. An unfeasible amount of time and effort would be required to mount the discs with such precision. Plastic deformation is likely to result from the mechanical loads induced by very heavy braking applications, its form taking that of the displacement graphs already shown for braking loads. This deformation would probably accrue over time. In both these instances one whole axial oscillation of the disc faces position at the pad location

would be experienced per disc revolution. Because of the combination of large clamping loads and high value of friction material Young's Modulus (this also provides a good braking response) any small variation of disc position as it passes through the caliper will result in a high rise in pad pressure.

The exact relationship between runout and pad pressure variation is not known and depends on several complex factors, cylinder friction, hydraulic system inertia and caliper sliding friction to name a few. It is assumed that the friction and inertia of the hydraulic system results in the pads being forced out to a mean position that is held roughly constant through the stop, with the variation of disc position being taken up by increases and decreases in pad compression and therefore contact pressure. It has also been shown that the pad compressibility is such that typical runout values of about 0.07mm for example are far too large for this to be the sole mechanism of pressure variation. If the pad were held rigidly at a 'mean' position the contact pressure would vary between vastly increased values and complete non contact of the disc and pad. It is therefore likely that some caliper movement takes place, sliding on the mounting. The forces required to oscillate the mass of the caliper with the rotating disc are not sufficient to contribute greatly to the pressure variation on the disc (this was analysed with simple vibration theory). The exact details are not known but sufficient clues are evident to suggest that the pressure variation is caused by varying pad compression and caliper oscillation as the brake disc rotates. This pressure variation leads to an equivalent temperature variation with increased temperatures in the regions of maximum runout. This was confirmed by experimental work completed at Ford (see Appendix VI). This temperature variation results in a coning variation around the disc (waving) which accentuates the original deformation. The thermal waving study showed that little heat variation is required for the effect to become self-exciting, a small initial wave or runout being magnified through subsequent brake applications. Softer friction pad material and good caliper sliding properties would help to slow down this build up and the rate and extent to which this deformation accrued would be dependent on the style of driving and the age of the disc.

However as disc mounting will never be perfect, even if the disc is rigid enough for virtually no plastic deformation to occur, this type of build up however slow may be very difficult to prevent in the first place. Therefore it then becomes important to limit the extent of thermal deformation so that unacceptable levels of waving are not

achieved. This limitation can be simplified to the prevention of coning, as it is a variation of this quantity around the disc which is waving itself.

Coning is caused by the radial expansion of the braking faces as they are heated. The disc hub, remains cool and does not expand, therefore deformation occurs in the disc section between the two as the rotor cheeks try to pull the disc wall outwards (Abbas et al.[6]). The resisting force exerted by the wall on the disc face to which it is attached results in there being an uneven expansion of the two faces, the unrestrained one pushing its opposite into a cone shape. The direction of coning is dependent on the orientation of the restrained and unrestrained disc faces and therefore is reversed in a conventional disc compared to an 'inside-out' disc. The extent of coning is governed by the resistance to expansion and the expansion forces themselves, in turn, governed by the disc section and the thickness of the disc rotor cheeks. These have been discussed and a hypothesis on the causes of coning is included earlier in this report.

Several approaches to the reduction of coning are presented in this thesis. Two main paths can be followed. Firstly the disc section can be increased, making the disc heavier, so that the disc is not allowed to expand and therefore deform. However, as the forces of thermal expansion are so great, large stress concentrations are developed in the disc and these would lead to cracking and plastic deformation, causing the build up of waving to occur even faster. Again a self-exciting situation would develop where stress levels would very quickly become unacceptable.

The second approach is to allow the disc to expand. It has been shown that reducing the resistance to expansion reduces the coning and as radial expansion on its own has no bearing on brake performance this is preferable. An extension of this idea is to design the disc so that the inevitable imbalance caused by only one braking face being restrained is counteracted by the uneven expansion caused by the disc faces being of different thicknesses, and yet being subject to similar flux inputs. Models have demonstrated this to be successful. The advantage of this method above a 'rigid' method such as the extension of the rib to the disc wall (Section 5.10) is that the uneven counterbalancing expansions vary in the same way as the restraining force of the wall section, as the magnitude of both depends directly on disc expansion. The 'uneven thickness' disc has been shown to yield good results for a variety of loading conditions.

Disc cooling is affected principally by the type of disc design. Conventional discs cool up to 25% faster than 'inside-out' because of the shorter path for heat to travel to the cool disc hub and because air is able to flow more easily through the ventilating passages in the disc. However the 'inside-out' disc design is inherently better at reducing coning and so the choice of design depends on the circumstances. This is discussed in more detail later. Because poor cooling is a function of the fundamental section design of an 'inside-out' disc it is preferable to try and improve the design of the conventional disc to cope with coning and retain the favourable cooling ability.

Therefore a disc with reduced susceptibility to thermal coning will result in a reduction of waving, the cause of the vast majority of waving deformation being variation of thermal inputs to the braking face around the disc.

It is important to consider the general form of this waving. Peaks and troughs in disc displacement are likely to cause hot spots, or regions, on the braking face in addition to heat rings. Heat rings are caused by localised pressure on the disc face, this in turn can be caused by poor caliper/pad design or localised thermal deformation during the brake application, resulting in a continual raised rubbing pressure at a particular radial position. Heat spots have also been observed, usually during dynamometer testing, but they have a tendency to move around the disc face. Often the appearance of a heat spot appears to be cyclic, it appears, grows, fades and disappears again. No relationship has been observed so far between heat spot position and fixed points on the disc, for example the regions of maximum runout.

The development of plastic deformation waving and waving effects in general will be occurring continuously and at arbitrary positions on the disc. Therefore as certain waves are generated during specific brake applications then many waves are coexisting and superposition would take place. This means that only the first 'perfect' brake application can be modelled and after that the increased number of waves, all combining and interfering with one another produces a very complex situation that would certainly be impossible to model. If, for the sake of simplicity, the deformation waves were taken to be sinusoidal, then the growth and decline of many waves of similar frequency over the top each other would produce the appearance of peaks and troughs, appearing as hot spots, growing and fading in apparently arbitrary places. This may be a somewhat simplistic approach but it is intended to illustrate the fact that hot spots tend to have varying positions because they are simply constructive additions

of possibly several moving waves at any given instant over a short period of time and that the position of hot spots will be changing continually throughout the disc life.

The fact that the complexity of these waves makes them virtually impossible to model accurately is not of great significance. If it can be established that initial waves are developed then it is the reduction of the coning that causes any wave to occur that is important. If the disc is not inclined to cone, the waving will not be so severe and the resulting hot spots caused by constructive interference will be reduced.

Other types of distortion are evident. In this case the outside edges of the rotor faces have a tendency to curl towards one another and this can be rectified by the extension of the ribs to the outside diameter of the disc. This also improves the uniform absorption of the heat into the disc, particularly in cases such as this one where a poor pad pressure distribution is evident.

6.2 'Inside-out' vs. Conventional?

During the course of this work several indications of the relative performance of the two types of disc have emerged. Initial segment tests confirmed that the section design of the 'inside-out' disc was far better at coping with coning. There were two main reasons. Firstly the effective wall length is longer and so less stiff, and less resistance to expansion is offered. Secondly the uneven expansion occurs in such a direction as to oppose the change in angle of the shoulder, instead of complement it as is the case with the conventional disc. The improvement in coning of the 'inside-out' versus conventional is about 50%.

However, the cooling tests that have been performed have confirmed the results obtained in practice, where the conventional disc cools approximately 25% faster than the 'inside-out' disc. The reason for this, as previously explained, are twofold. The disc section of the conventional disc is such that a much shorter thermal path is available to heat flowing to the cool hub section of the disc. It therefore cools by conduction more effectively. In addition the path for air flowing through the ventilating passages is much more straightforward and so even though convective face cooling is the same in both discs, the conventional disc is more effective as a ventilated disc, heat being lost in the air passages again more easily.

In addition to these factors, the 'inside-out' disc appears to be stiffer when subjected to mechanical pad loads. If plastic deformation as a result of these loads is at the heart of waving then the 'inside-out' disc will have an advantage over the conventional. Shoulder stresses are higher during coning however.

Unfortunately, there is no simple answer to which is the 'best' design of disc. It depends on the projected usage of the disc. If a disc is required that has good deformation behaviour, for instance a disc that is likely to be used with infrequent but heavy applications, then the 'inside-out' disc is better. Conversely a disc that has a high rate of application, but usually less demanding, then the conventional disc is preferable. This decision even is not clear cut and the performance of particular aspects of the disc can be enhanced as desired (sometimes to the detriment of other aspects) using the information contained in this report. It still does remain the choice of the designer as to what type of disc performance is required, although hopefully his/her job of designing a disc to conform to these requirements will be made easier by the findings of this research. It is, of course, more desirable to achieve both characteristics of good cooling and distortion performance in a single design of disc and this is attempted, using information summarised at the beginning of this Section, in the remainder of this Chapter.

6.3 Improved Disc Design

6.3.1 Design Procedure

The procedure for the design of an improved disc is given below. This was attempted at the end of the experimental work and is intended as an illustration of the use of some of the design guidelines.

The first step was to define the requirements of the disc. The most important criterion was decided to be coning, and its ensuing effects on thermal waving and ultimately judder performance of the vehicle. This was chosen as of primary importance because in the initial stages of the project there was great concern about the occurrence of this phenomenon in Ford customer vehicles and one of the aims of the work was to try and produce some insight into it. Following this cooling was considered vital. As previously discussed, the inherent section of the 'inside-out' disc causes its inferior

cooling and so it was decided to use a conventional design of disc, it being easier to reduce the coning on a conventional disc than to make an 'inside-out' disc cool faster.

This having been established it is worth restating the criteria that any revised design of disc should remain roughly within the envelope of the current disc. Therefore the disc diameter was necessarily kept the same. Under conditions where no limits are placed on disc packaging the general sizing of the disc is best accomplished using methods such as those described by Sheridan, Kutchey and Samie [19] (in this case it is also worth devoting considerable time to creating a pad/caliper design with a near uniform pressure distribution). It is also necessary to design the disc with the current pad pressure distribution borne in mind.

The main feature of the disc to be altered was the face thickness. Face #1 remained at 7mm whilst face #2 was thickened to 12mm. This produced an increase in weight, but it was felt that this could be justified by the improved performance. Naturally, reducing the thickness of face #1 should produce the same effect reducing coning but it was felt that the current stress levels should be reduced and not exceeded. It was also decided to vary the rib width over the face of the disc and a variation of 6mm at the inside edge to 10mm at the outside edge was selected. The ribs were also extended to the outside diameter with the aim of curbing the curling of the braking faces that was taking place at the outside edge. As far as the middle section of the disc was concerned, the curved shoulder was implemented and the depth of the disc increased by 6mm (this increase in depth should not adversely affect disc packaging as it effectively moves the caliper slightly away from the wheel where space is most restricted). This combination had been shown to reduce the stiffness counteracting expansion and therefore coning. An undercut was not included as, in a ventilated disc (especially with a curved shoulder) the benefits were low. An illustration of the improved segment model is shown in Fig 47. Models of the segment and the full disc were created and the standard single stop inputs and braking loads applied to them. Cooling, waving and drag tests were all applied in addition and the results displayed below.

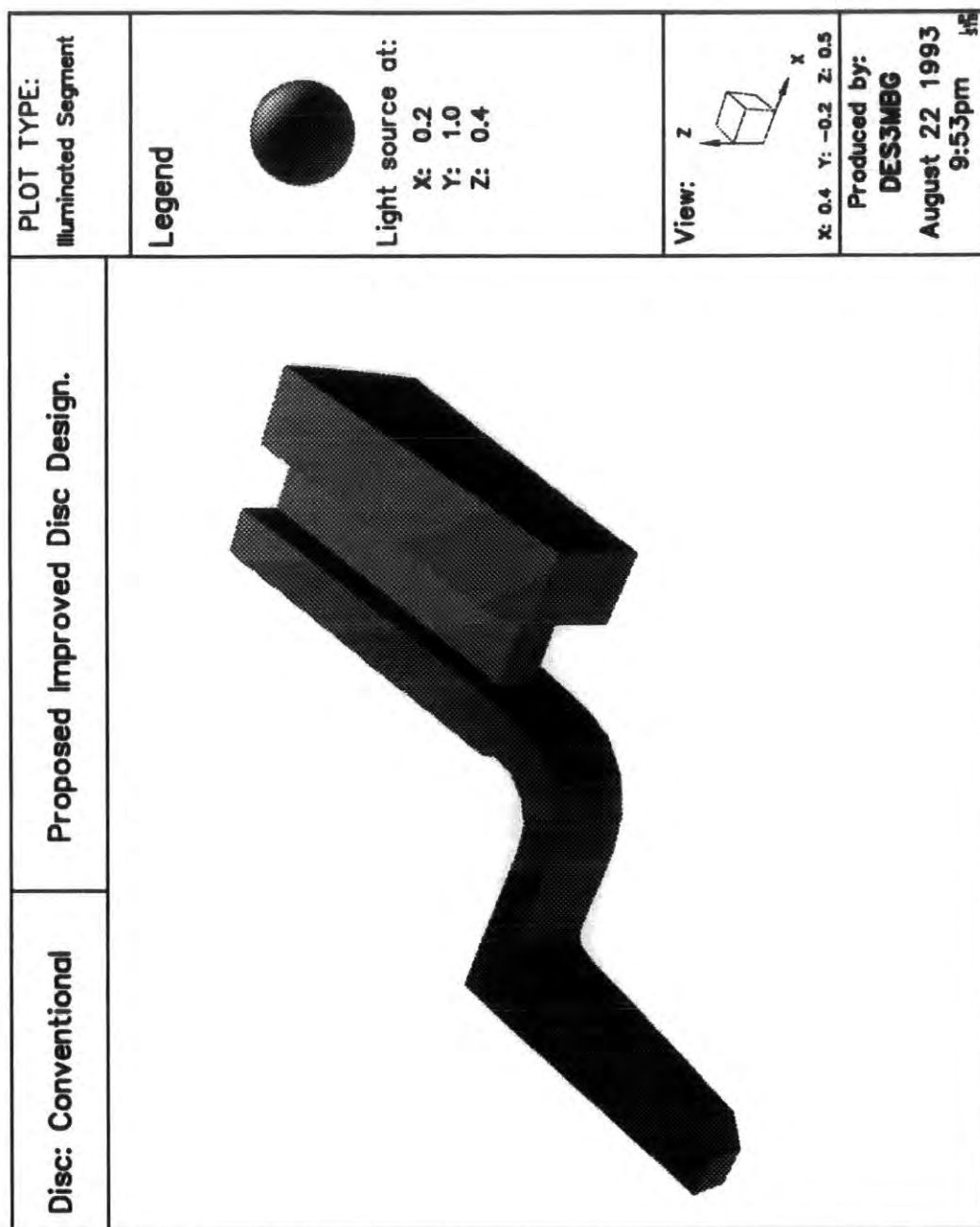


Fig 47. Disc segment - proposed design.

6.3.2 Results & Analysis

1. Single Stop Test

The following figures show a comparison of 'inside-out' design, conventional design and the proposed design for temperature, displacement and stress conditions. The summary table shows the relative values.

Quantity	Braking Face	'Inside-out'	Conventional	Proposed
Max. Face Temperature	1	849.2	849.5	712.3
	2	760.3	773.4	499.2
Mean Face Temperature	1	706.5	643.8	553.3
	2	632.6	701.8	462.1
Shoulder Temperature	N/A	119.3	93.6	N/A

Table 26. Temperature summary - proposed design.

Quantity	Braking Face/Direction	'Inside-out'	Conventional	Proposed
Coning Angle (deg.)	1	0.48	-0.99	-0.13
	2	0.53	-0.86	0.00
Edge Deflection (mm)	1	0.17	-1.33	-0.35
	2	0.40	-1.10	-0.16
Radial Expansion (mm)	1	1.21	0.94	0.86
	2	0.98	1.27	0.78
Shoulder Displacement (mm)	X	0.04	0.07	N/A
	Y	0.50	0.49	N/A

Table 27. Displacement summary - proposed design.

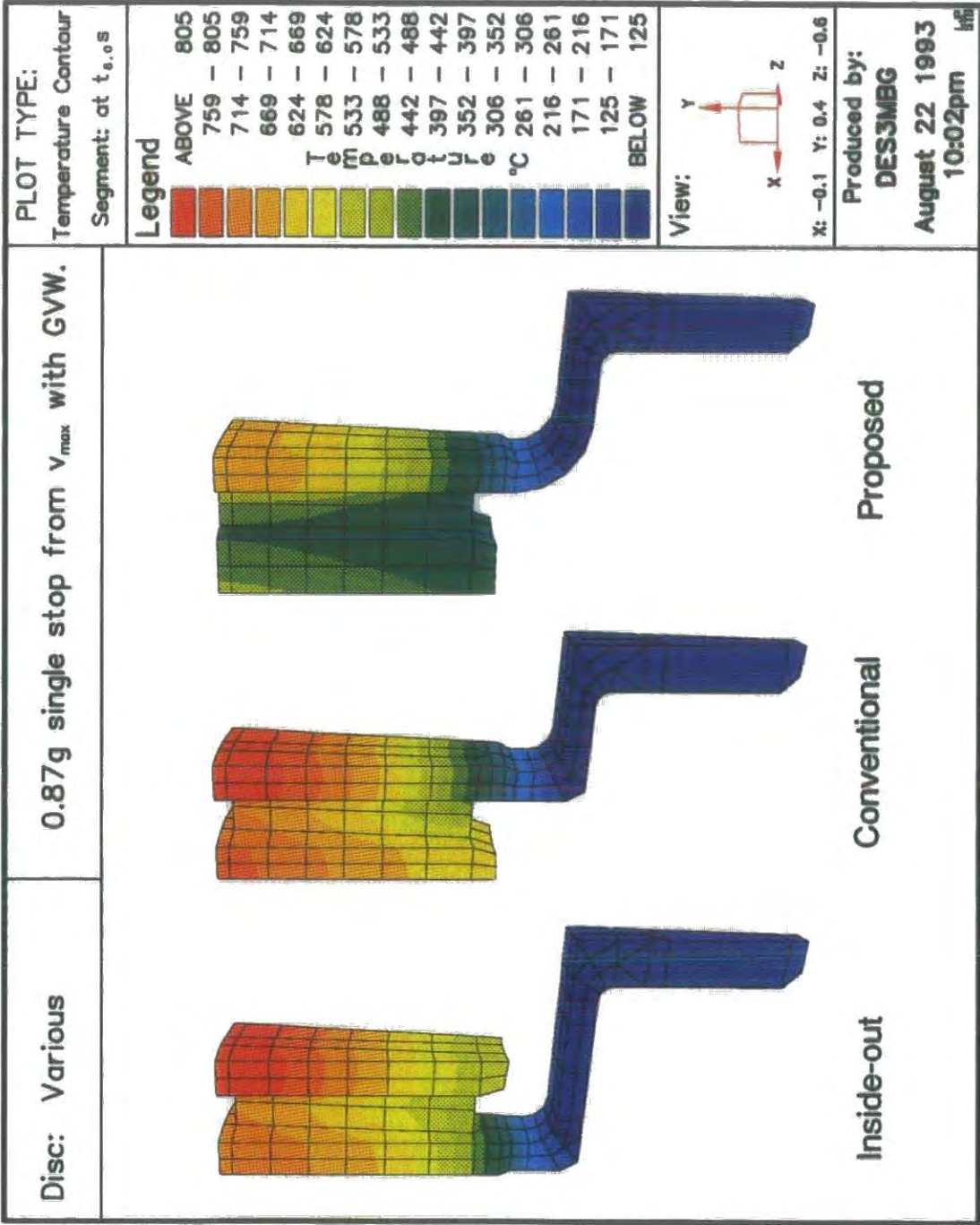


Fig 48. Temperature distribution - proposed design.

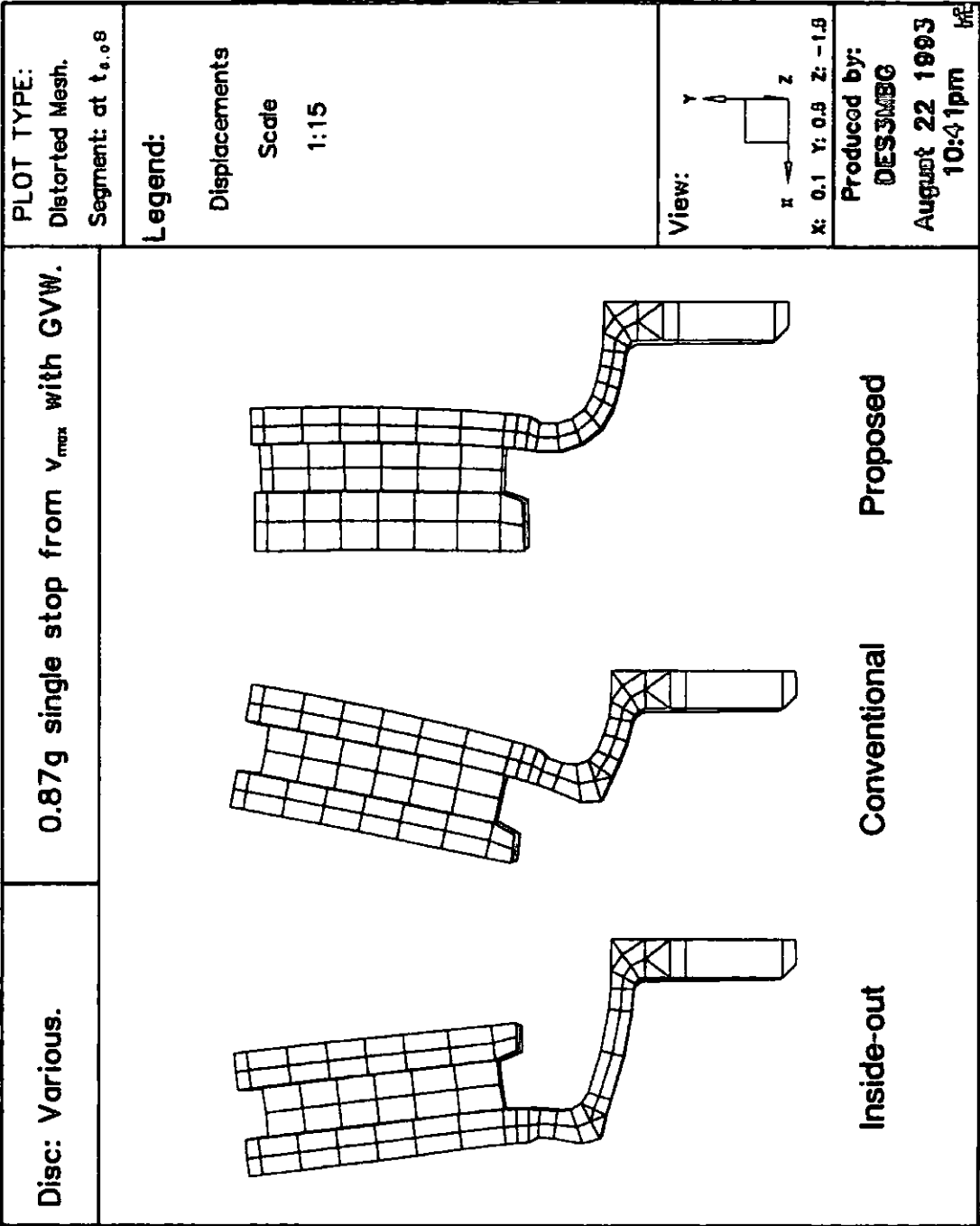
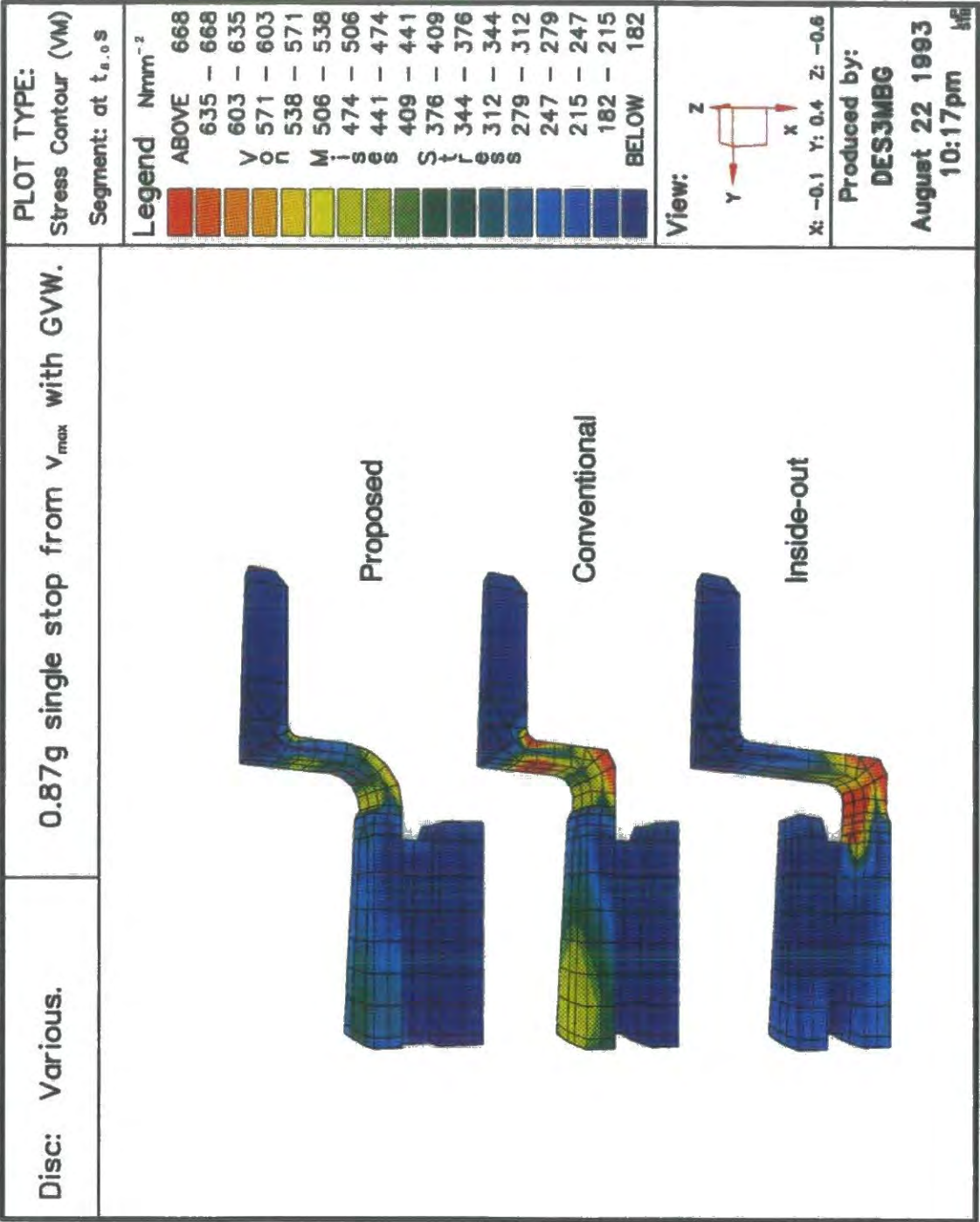


Fig 49. Displaced shape - proposed design.



2. Drag Test

Tables 28 and 29 show the results obtained from the drag test. The proposed design does not perform as well as the 'inside-out' as far as coning is concerned, but it is an improvement on the conventional design. In this way the improved coning and cooling properties of the two types of discs have been incorporated in the proposed design.

Quantity	Braking Face	'Inside-out'	Conventional	Proposed
Max. Face	1	403.1	356.1	368.6
Temperature	2	398.5	356.2	367.1
Mean Face	1	391.9	323.1	341.2
Temperature	2	373.3	345.3	360.1
Shoulder	N/A	262.4	193.0	N/A
Temperature				

Table 28. Drag test temperature results - proposed disc design.

Quantity	Braking Face/Direction	'Inside-out'	Conventional	Proposed
Coning Angle (deg.)	1	0.10	-0.36	-0.23
	2	0.10	-0.32	-0.19
Edge Deflection (mm)	1	-0.01	-0.49	-0.32
	2	0.11	-0.38	-0.19
Radial Expansion (mm)	1	0.67	0.50	0.55
	2	0.63	0.63	0.65
Shoulder	X	0.09	0.06	N/A
Displacement (mm)	Y	0.32	0.25	N/A

Table 29. Drag test displacement results - proposed disc design.

3. Mechanical Load Deflections

The thicker face proves advantageous in reducing the amplitude of waving caused by pure mechanical loads. This increase in stiffness should contribute towards a slower build up of thermal waving. The graph illustrating the waving is shown below.

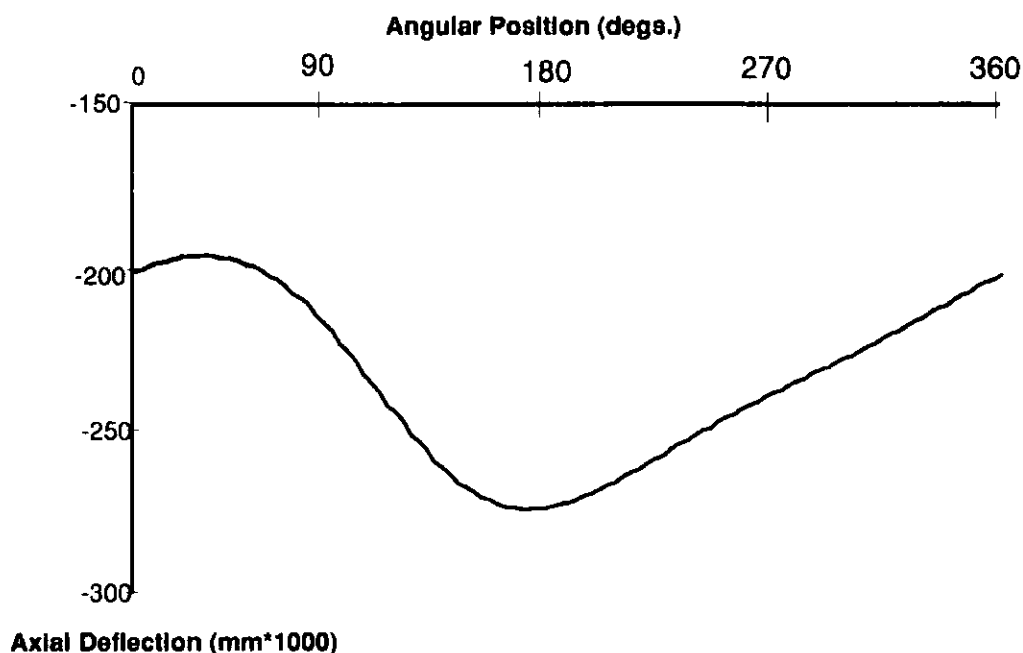


Fig 51. Axial deflection due to friction loads - proposed design.

4. Disc Mass Differences

The proposed design is 20% heavier than the current 'inside-out' design. This need not be the case. One face can be reduced to create the ratio between the two faces resulting in reduced coning, but in this case it was felt that as a reduction in thickness would increase the temperatures on that face, increasing stresses in the shoulder that a much improved design of disc would be achieved with a small mass increase. This is entirely up to the designer. If the priorities are recognised as being mass over coning or cracking then alternative routes can be taken.

CHAPTER 7

CONCLUSIONS

7.1 Disc Design

Although much of this work has yet to be put into practice it is hoped that the reasons for and the mechanisms of brake disc thermal distortion have been demonstrated with clarity. Disc design will never be a rigidly defined procedure, the judgement and experience of the designer are naturally crucial but it is necessary to found judgements on sound bases and it is to this end that this thesis has been prepared. It is important to design the disc for the uses to which it will be put and it is hoped that adequate demonstration has been made of the possibility for producing a design with quite particular performance in mind. Many variables are there to be changed and if some understanding is obtained of the effect of each then a disc can be designed with very specific aims in mind and with as little mass as possible.

Although the research documented in this report consistently used a particular disc as an example, the results and trends obtained should have broad agreement with the response of any similar design of ventilated brake disc. In this case there appears to be a more favourable disc response with the proposed disc design than with the original, the main advantage over the current, 'inside-out' disc being its improved cooling ability. It is of course necessary to provide some experimental evidence of its benefits and it is hoped that Ford will take up the recommendations and produce some prototypes for evaluation. In the case of the 'improved' design there is an increase in disc mass. It would be possible, by reducing the face thicknesses whilst keeping the

approximate ratio the same, to maintain excellent coning performance but with a lighter disc. However in the example given, it was felt that the risk of worsening the waving by reduction of disc stiffness outweighed the disadvantages of a slightly heavier disc. This again is the choice of the designer.

It is hoped that as the current packaging as been adhered to reasonably faithfully that the cost of implementation of the disc, including manufacturing changes and other local design changes, should be as low as is possible for the introduction of any new component. Extensive testing would be required, the component having such a fundamental affect on the safety, performance and ride quality of the vehicle.

It is worth concluding that although excellent results have been achieved in computer simulation that the application of the exact dimensions suggested here may not be an optimum. As with any component designed by mathematical modelling, prototype development and testing is hopefully significantly reduced but it is not removed altogether. If the changed design is to be used then some experimentation with the exact values is still necessary to ascertain effectiveness and optimum conditions. As has already been said, this optimum cannot be produced solely on the computer as it is readily admitted that, although there is great confidence in the principles and the effects of lowering this dimension or extending that one, it is not possible to predict with fine accuracy exactly the response the disc will have in use. This is not a disclaimer against any inaccuracy but a simple statement of a practical fact with this type of testing. All efforts in the modelling have been made to ensure that results are as accurate as possible, consistent with the standard of information requested from the research and the time available. The comparative testing and the test conditions themselves were all selected with this aim in mind.

7.2 Caliper/Pad Design

It was not originally within the brief of this project to examine these components but the results of the necessary modelling of the pads are important and need discussing. As has been said many times previously, the pad pressure distribution is important to achieving stable braking conditions. There is little point in designing a disc with great accuracy if the loads placed upon it are detrimental to its operation. In the case of the pads examined here, there is a vast difference between the two, caused by necessary

caliper design and a combination of high friction pad material compressibility and chosen pad backplate thickness. Pad #1 in particular is very bad, the pressure concentration occurring at the outer edge of the disc. The greater disc surface speed at this point combined with this increased pressure will result in an even greater heat input variation across the braking face (the outside being the worst place to have a large heat input as far as effective heat loss to the hub is concerned) and the resulting pad wear will be more severe. In addition, and this was not modelled, any caliper flexure during brake application will tend to have the effect of *increasing* this localised high contact pressure. It is accepted that caliper flexure can not be designed away entirely, and the poor pressure distribution is caused by the design of the non-piston side of the caliper. This design exists because of manufacturing limitations and so changes to caliper design would prove difficult and costly. However, an increase in pad backplate thickness would be relatively inexpensive and would have large benefits as far as the pressure distribution was concerned (it would also prove more simple than the development and subsequent wear testing of a new softer friction material). In this way very similar distributions could be obtained for both sides of the disc and effects such as the mechanical coning caused by the mis-alignment of the pad centres of pressure could be avoided. Only a small amount of work was devoted to the study of pad design in this project but it is worth ensuring good performance from this inexpensive and relatively simple component before the disc is designed.

It is also worth briefly mentioning that the careful design of suspension components so as not to allow longitudinal oscillation of the axle, particularly ensuring the natural frequency is not in the likely range of wheel rotations, should not be overlooked. This can be ignored and unreasonable requirements placed on disc performance when it is perceived as the sole cause of any vibrations.

7.3 Method Appraisal

The modelling of the discs and the generation of the simulation conditions and the data processing programs took by far the majority of the time available for the project. Development of the model is naturally crucial to the accuracy of the work, and so it is worth spending much time on it. Once the model exists it is relatively simple to make geometry changes and perform variation analyses. This part of the experimental work formed a very small proportion of the total time! Although seemingly time

consuming, this method highlights the critical features of the disc affecting its performance more quickly and with less investment than with a programme of prototype brake disc manufacture and testing. This vastly reduces prototype component lead time and cost of realising an improved design.

In order to produce more accurate results it would be necessary to conduct a significant amount of investigation into the exact air flow conditions existing around the disc in addition to the friction layer and its behaviour. Detailed work is being undertaken in this field at the moment and much is expected to be discussed in the coming years. As in most simulations, the majority of the inaccuracy will derive from the assumptions made about thermal inputs. These were carefully evaluated using evidence found by other researchers and the validations described in Section 4. It is necessary to define the level of accuracy being aimed for within the context of the project brief and the strict use of comparative testing in this project (no *exact* predictions about in-service behaviour but comparisons with the current disc design) ensured that the results were sufficiently accurate to justify investment into disc re-design and testing.

The computer programs used for the processing of data were all checked rigorously and no known significant source of error exists within these. The model meshes themselves were all faithful reproductions of the actual designs they represented.

7.4 Further Work

There is plenty of potential for work to be continued using the model and pre/post-processing systems established for this work. Dimension variations can be modelled and refined in greater and greater detail but little further work to produce information on general disc behaviour is likely. The next step would be to employ greater computing power, and much more complex solution method, certainly incorporating an iterative scheme, and study disc effects in more detail. However the extra investment required to significantly improve accuracy could only be justified by detailed investigations into such characteristics as friction layer deposition and the macroscopic material changes that are undergone to bring it about. As far as general disc behaviour is concerned, I believe there is little to be gained by significant increases in simulation complexity.

The use of thermal imaging equipment in conjunction with a Finite Element Analysis is a very positive step toward accurate and efficient use of numerical solutions. Providing the data transfer is possible between the experimental apparatus and the computer system, measured thermal values can be imposed on the mesh. This removes the largest source of error in the simulation, namely the assumptions made about the thermal inputs to the disc. If temperature maps can easily be placed into the modelling system then the known conditions can be used as the standard input. The model can then be changed freely and the power of finite element modelling used to the full. The numerical solution side of the modelling can be extremely accurate and by vastly reducing the main source of error even more value can be gained. This is almost certainly the way forward for analyses such as this one, involving brake discs and its success is dependent on the ability to transfer the thermal imaging data to the model effectively.

Disc design, as with practically all other component design, is a matter of compromise. It requires performance objectives and priorities to be recognised and choices about the design to be made according to its defined specification. In order to be as informed as possible when make these choices it is necessary for the brake designer to have an understanding of the behaviour of the disc under its operating conditions and the effects of the numerous variables within its design. It is toward the understanding of these underlying principles that the work described within this thesis has been aimed and it is hoped that design guidelines and a method have been demonstrated which could make disc design significantly more deliberate and efficient.

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APPENDIX I

DISC/PAD DATA

The following Appendix provides details of the disc and pads used in the simulations.

A1.1 Disc

The brake disc used as the basis for the modelling was the 'inside-out' disc used on certain derivatives of the Ford Mondeo. Some of its dimensions are shown in the diagram below.

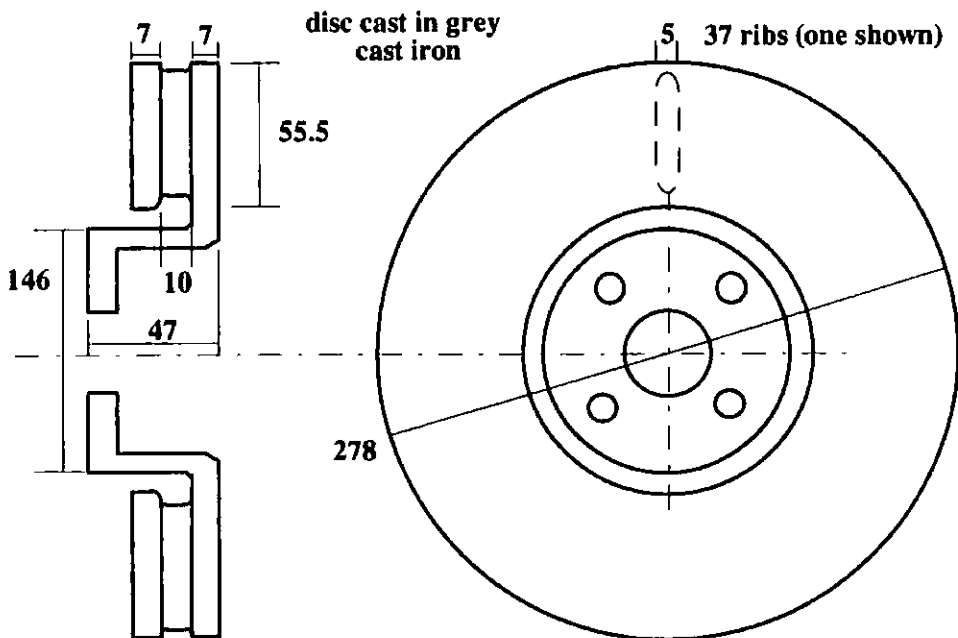


Fig A11 Important disc dimensions.

Disc material properties used in the analysis are given below.

Grey Cast Iron

Young's Modulus	Poisson's Ratio	Mass Density	Coefficient of Thermal Expansion	Hysteretic Damping	Thermal Conduct.	Specific Heat
Nmm ⁻²	-	kg mm ⁻³	°C ⁻¹	-	Jm ⁻¹ s ⁻¹ °C ⁻¹	Jkg ⁻¹ °C ⁻¹
125E3	0.25	7.15E-6	12.5E-6	0.005	46E-3	505

A1.2 Pad

The pad was of the chamfered type and included lugs to locate it within the caliper. Pad dimensions are listed below.

Backplate thickness:	5.5mm
Friction material depth:	13mm
Pad contact area:	4300mm ²

The backplate was manufactured from mild steel, and the friction material was of a Teves GmbH specification. Material properties, provided by the manufacturer, are listed below (friction material properties in bold text).

Young's Modulus	Poisson's Ratio	Mass Density	Coefficient of Thermal Expansion	Hysteretic Damping	Thermal Conduct.	Specific Heat
Nmm ⁻²	-	kg mm ⁻³	°C ⁻¹	-	Jm ⁻¹ s ⁻¹ °C ⁻¹	Jkg ⁻¹ °C ⁻¹
206.0E3	0.35	7.80E-6	11.0E-6	0.005	41.50E-3	480
9.0E3	0.30	2.50E-6	14.3E-6	0.050	2.06E-3	749

The coefficient of friction used for the pad material was 0.4.

The friction material compressibility in Section 5.7 was varied to obtain a 'hard' and a 'soft' pad material. For the 'hard' material, Young's Modulus was increased to 9.0E4Nmm⁻² and for the 'soft' it was decreased to 9.0E2Nmm⁻².

APPENDIX II

EXAMPLE MODEL FILES

The following Appendix contains two example model files. They are the reference models ('inside-out') for both temperature and stress calculations.

The first model calculates a temperature distribution for the mesh throughout the braking stop. It uses the flux inputs near the end of the listing, as calculated from the pad contact pressure model.

Temperature Calculation Model:

```
TITLE  INSIDE-OUT DISC - TEMPERATURE REFERENCE MODEL
C      THERMAL DISTRIBUTION
C
C      M.B. GERRARD
C      APPLIED MECHANICS GROUP
C      S.E.C.S.
C      UNIVERSITY OF DURHAM
C
C
C
CONTROL
CALC.STEADY.TEMPS
CALC.TRANS.TEMPS
SAVE.TEMPS
FULL.CONTROL
PHASE=1
TOLERANCE=10E-2
PHASE=2
TOLERANCE=10E-2
PHASE=4
PHASE=6
```

PHASE=7
CONTROL.END
NODES
AXIS.NUMBER=2
NODE.NUMBER X Y Z
1,0.00,0.00,0.00
2,5.50,31.75,0.00
3,0.00,31.75,0.00
4,8.50,34.75,0.00
5,0.00,34.75,0.00
6,8.50,56.40,0.00
7,0.00,56.40,0.00
8,8.50,59.83,0.00
9,0.00,59.83,0.00
10,8.50,65.50,0.00
11,4.25,65.50,0.00

Majority of node co-ordinate list omitted.

178,2.00,2.00,0.00
179,2.00,2.00,4.86
180,38.03,70.84,0.00
181,38.56,68.00,0.00
182,38.03,70.84,4.86
183,38.56,68.00,4.86
PAFBLOCKS
PROPERTIES=11
TYPE ELEMENT TYPE N1 N2 N3 N4 N5 TOPOLOGY
1 39710 1 1 0 0 1 2 3 202 203 4 5 204 205
1 39710 1 1 0 0 1 4 5 204 205 6 7 206 207
3 39610 2 1 0 0 1 208 8 209 9 206 6 207 7
3 39610 2 1 0 0 2 10 12 210 212 8 9 208 209
1 39710 1 1 0 0 2 11 10 211 210 17 13 217 213
1 39710 1 1 0 0 2 12 11 212 211 18 17 218 217
2 39610 1 1 1 0 2 19 12 18 219 212 218

Majority of element topology list omitted.

1 39710 1 1 0 0 2 17 13 217 213 21 16 221 216
1 39710 2 1 0 0 1 168 169 344 345 170 171 346 347
1 39710 2 1 0 0 1 173 172 175 174 136 137 148 149
1 39710 2 1 0 0 1 175 174 177 176 148 149 160 161
1 39710 2 1 0 0 1 177 176 373 372 160 161 336 337
MATERIAL
MATERIAL.NUMBER E NU RO ALPHA MU K SH
11 125E3 0.25 7.15E-06 12.5E-06 0.005 46E-3 505
MESH
REFERENCE SPACING.LIST
1 1
2 2
3 0.6 1.2 1.2


```

AXES
AXISNO RELAXISNO TYPE NODE.NO ANG3
4 2 2 1 9.73
SIMILAR.NODES
ORIGINAL.NODE=1
NUMBER.OF.NODES=183
COPY.NODE AXIS.OF.NEW.NODES
201 4
TEMPERATURE
TEMPERATURE LIST.OF.NODES
20 2
UNSTEADY.THERMAL.TIMES
TIME.STEP MAX.TIME NUMBER
0.25 8.0 8
HEAT.TRANSFER
TYPE.OF.HEAT.TRANSFER AMBIENT FILM PLANE AXIS N1
1 20 1 1 2 4
1 20 1 1 2 5
NODAL.FLUX.SHOCK
NODE.NUMBER FLUX.TIME.LIST

```

```

47 -.653E+01 .00 -.631E+01 .25 -.609E+01 .50 -.586E+01 .75
* -.564E+01 1.00 -.542E+01 1.25 -.520E+01 1.50 -.498E+01 1.75
* -.476E+01 2.00 -.454E+01 2.25 -.432E+01 2.50 -.409E+01 2.75
* -.387E+01 3.00 -.365E+01 3.25 -.343E+01 3.50 -.321E+01 3.75
* -.299E+01 4.00 -.277E+01 4.25 -.254E+01 4.50 -.232E+01 4.75 -.210E+01 5.00
* -.188E+01 5.25 -.166E+01 5.50 -.144E+01 5.75 -.122E+01 6.00 -.996E+00 6.25
* -.775E+00 6.50 -.553E+00 6.75 -.332E+00 7.00 -.111E+00 7.25 .000E+00 7.50
50 -.131E+02 .00 -.126E+02 .25 -.122E+02 .50 -.117E+02 .75
* -.113E+02 1.00 -.108E+02 1.25 -.104E+02 1.50 -.996E+01 1.75
* -.952E+01 2.00 -.907E+01 2.25 -.863E+01 2.50 -.819E+01 2.75
* -.775E+01 3.00 -.730E+01 3.25 -.686E+01 3.50 -.642E+01 3.75
* -.597E+01 4.00 -.553E+01 4.25 -.509E+01 4.50 -.465E+01 4.75 -.420E+01 5.00
* -.376E+01 5.25 -.332E+01 5.50 -.288E+01 5.75 -.243E+01 6.00 -.199E+01 6.25
* -.155E+01 6.50 -.111E+01 6.75 -.664E+00 7.00 -.221E+00 7.25 .000E+00 7.50

```

Each block contains a description of the flux input at a single node for
each of the 0.25s time steps throughout the simulation.

The majority have been removed.

```

1402 .479E+02 .00 .463E+02 .25 .447E+02 .50 .431E+02 .75
* .414E+02 1.00 .398E+02 1.25 .382E+02 1.50 .366E+02 1.75
* .349E+02 2.00 .333E+02 2.25 .317E+02 2.50 .301E+02 2.75
* .284E+02 3.00 .268E+02 3.25 .252E+02 3.50 .236E+02 3.75
* .219E+02 4.00 .203E+02 4.25 .187E+02 4.50 .171E+02 4.75 .154E+02 5.00
* .138E+02 5.25 .122E+02 5.50 .106E+02 5.75 .894E+01 6.00 .731E+01 6.25
* .569E+01 6.50 .406E+01 6.75 .244E+01 7.00 .813E+00 7.25 .000E+00 7.50
END.OF.DATA

```

The following model file uses the temperature distribution calculated by the previous model to derive thermal strains and stresses, in addition to any deformation caused by mechanical loads.

Displacement Calculation Model:

```
TITLE  INSIDE-OUT DISC - STRESS/DISPLACEMENT REFERENCE MODEL
C      DEFORMED SHAPE
C
C      M.B. GERRARD
C      APPLIED MECHANICS GROUP
C      S.E.C.S.
C      UNIVERSITY OF DURHAM
C
C
C
CONTROL
READ.TEMPS.FROM.93BBTMP.ST
FULL.CONTROL
PHASE=1
TOLERANCE=10E-2
PHASE=2
TOLERANCE=10E-2
PHASE=4
TOLERANCE=10E-2
PHASE=6
PHASE=7
PHASE=9
CONTROL.END
NODES
AXIS.NUMBER=2
NODE.NUMBER X Y Z
1,0.00,0.00,0.00
2,5.50,31.75,0.00
3,0.00,31.75,0.00
4,8.50,34.75,0.00
5,0.00,34.75,0.00
6,8.50,56.40,0.00
7,0.00,56.40,0.00
8,8.50,59.83,0.00
9,0.00,59.83,0.00
10,8.50,65.50,0.00
11,4.25,65.50,0.00
12,0.00,65.50,0.009
```

=====

Majority of node co-ordinate list omitted.

=====

```
177,28.55,84.00,6.68
178,2.00,2.00,0.00
```

179,2.00,2.00,4.86
 180,38.03,70.84,0.00
 181,38.56,68.00,0.00
 182,38.03,70.84,4.86
 183,38.56,68.00,4.86

PAFBLOCKS

PROPERTIES=11

TYPE ELEMENT TYPE N1 N2 N3 N4 N5 TOPOLOGY

1	37110	1	1	0	0	1	2	3	202	203	4	5	204	205
1	37110	1	1	0	0	1	4	5	204	205	6	7	206	207
3	37210	2	1	0	0	1	208	8	209	9	206	6	207	7
3	37210	2	1	0	0	2	10	12	210	212	8	9	208	209
1	37110	1	1	0	0	2	11	10	211	210	17	13	217	213

Majority of element topology list omitted.

1	37110	1	1	0	0	2	17	13	217	213	21	16	221	216
1	37110	2	1	0	0	1	168	169	344	345	170	171	346	347
1	37110	2	1	0	0	1	173	172	175	174	136	137	148	149
1	37110	2	1	0	0	1	175	174	177	176	148	149	160	161
1	37110	2	1	0	0	1	177	176	373	372	160	161	336	337

MATERIAL

MATERIAL.NUMBER E NU RO ALPHA MU K SH

11	125E3	0.25	7.15E-06	12.5E-06	0.005	46E-3	505
----	-------	------	----------	----------	-------	-------	-----

MESH

REFERENCE SPACING.LIST

1	1
2	2
3	0.6 1.2 1.2

AXES

AXISNO RELAXISNO TYPE NODE.NO ANG3

4	2	2	1	9.73
5	2	1	1	9.73

SIMILAR.NODES

ORIGINAL.NODE=1

NUMBER.OF.NODES=183

COPY.NODE AXIS.OF.NEW.NODES

201	4
-----	---

LOCAL.DIRECTIONS

NODE.NUMBER LOCAL.AXIS PLANE AXIS.NUMBER

204	5	3	2
-----	---	---	---

TIMES.FOR.THERMAL.STRESS.CALCULATION

TIME LOAD.CASE

2.0	1
4.0	2
6.0	3
8.0	4

RESTRAINTS

NODE.NUMBER PLANE AXIS.NUMBER DIRECTION

6	1	2	123
7	1	2	123
4	3	2	3
204	3	2	3

PROCESSING.FOR.PRINTED.OUTPUT
ORDER FORMAT.TYPE WINDOW
1 2 1
ORDER.FOR.PRINTED.OUTPUT
ORDER LIST.OF.TYPES
1 101 102 4 2 3
WINDOWS.FOR.PRINTED.OUTPUT
WINDOW GROUP
1 1
END.OF.DATA

APPENDIX III

LIST OF PROGRAMS

The following Appendix provides some details of the FORTRAN programs written to process data for the simulations. The filename and a brief description of each is given. In addition, similar reference is made to the spreadsheets used to generate the mesh topologies for the various different models.

A3.1 Data Processing Programs

Program Name	
---------------------	--

gphget1.f	These were 'get' programs, written to retrieve data from PAFEC output files and store it in a suitable format to be used by the graphics generation programs. The first two programs obtain geometric data from the node co-ordinates and element topologies for the creation of the solid models.
gphget2.f	
sstmpget.f	
ttmpget.f	
dispget.f	
strget.f	The last four concern temperature (steady-state and transient), displacement and stress values and once obtained these are superimposed on the solid models by the following group of programs.

A3.2 Graphics Generation Programs

The following programs all use the output of those above to generate the graphical output seen throughout this report.

Program Name

lightgen.f
tempgen.f
dispgen.f
strgen.f
multi.f
sectgen.f

'lightgen.f' was written to produce illuminated solid models of any model mesh, creating a more easily visualised object. 'tempgen.f' and 'strgen.f' produced coloured contour plots (temperature and stress respectively) and 'dispgen.f' produced displaced shape plots, again allowing the deformation of any mesh to be assimilated very quickly. 'multi.f' allowed a model to be viewed proceeding through the whole braking stop, a new contoured model being created for each time step. 'sectgen.f' allowed sections through the model to be selected and the temperatures or stresses viewed over that section.

A3.3 Results Summary Programs

Program Name

tempres.f
dispres.f

These programs retrieved specific values from the files created by the 'get' programs and used them to compile summary tables for temperature and displacement, very similar to those throughout Section 5. Analysis of new models could be completed more quickly and more accurately by repeating the same program for different model cases.

A3.4 Special Programs

The following two programs were created for specific tasks and are both very complex.

A program was required to convert the geometry of the segment model into a list of nodes and elements for a full disc model suitable for placing into a PAFEC model file. 'datagen.f' used the output node/element list produced by PAFEC and manipulated it, effectively using the cyclic symmetry of the disc, to create a new model file containing all the geometry for the full disc model. This would have been impossible to accomplish without such a program.

In addition it was necessary to write a program that converted the contact pressure distribution obtained from the pad model into a set of heat flux inputs for either a segment of full disc model. It retrieved stresses in a similar way to the 'get' programs from the PAFEC output file and using all the other factors involved, such as pad μ , disc surface speed, thermal partitioning ratio and others, it created a flux map over the surface of the disc braking face. This map had then to be interpolated and distributed across the nodes on the braking face of the model mesh and this was achieved with the help of UNIRAS mapping/interpolation routines. Value weighting techniques were also required. This program took much time to develop but once created its operation was straightforward and accurate. This program lies at the heart of the method for using the pad pressure distribution directly to calculate very accurate flux inputs into the disc and, again, without it this method would not have been possible.

A3.5 Model Generation Spreadsheets

Several spreadsheets were employed to create model meshes as explained in Section 3.1.3.

S/Sheet Name

93bbund.xls

93bbro.xls

93bbhat.xls

93bbrib.xls

93bbag.xls

The sheets listed all applied to the 'inside-out' design of model and there was an equivalent set of spreadsheets for the conventional design. Undercut variation, braking face thickness, disc depth, rib width and varying air gap models were created using the spreadsheets shown here. Each one was based on the general spreadsheet for the disc type and the required co-ordinate cells linked to a single cell denoting the dimension to be varied. The function linking the two was naturally dependent on the type of dimension and the position of the node.

APPENDIX IV

TEST CONDITIONS

The following conditions were agreed upon after discussions with Ford engineers. These conditions were adhered to in the computer modelling but were also used to run the dynamometer tests described in Appendix VI.

A4.1 Disc/Pad Conditioning

It was important to specify the initial condition of the disc and pads for dynamometer testing. The simplest and most accurate condition to model is that of complete contact between pad and disc at the start of the test. Because of this, 150 light brake applications prior to the dynamometer test were specified to ensure that the pads were fully bedded, ensuring the physical test conditions mirrored the simulations. Care was to be taken during these light brake applications during bedding to preserve pad material composition at the contact face, any excessive braking would impose thermal deformations on the disc and causes variations in pad μ before the main testing had begun.

A4.2 Test Conditions

The main conditions of testing were that a theoretical vehicle weight of 1875kg was to be used and that the deceleration to be modelled would be 0.87g. This brake application would commence at the maximum speed of the vehicle (v_{max} - 230kph). It was felt that these figures (the mass is a widely recognised Ford standard of vehicle

loading - GVW) were reasonable values for an emergency stop and yet would be suitably severe in order to produce large deformations. Large deformations and high temperatures were desirable as they would be easier and therefore more accurate to measure. Using component data, these conditions can be converted to braking torque (c2100Nm) and a brake line pressure (7.78Nmm^{-2}). These values were then used on the dynamometer.

It was also necessary to assume a constant pad friction coefficient value (see assumptions in Section 3.3.1) and 0.4 was agreed upon. In practice this may have been a little high but the prediction of this value is difficult because of its continual variation and its dependence on factors not able to be included in the computer model, e.g. friction film changes and pad material composition changes in the region of the pad face.

The disc and pads were specified as being at ambient temperature before the test.

Output was requested in the form of thermal image maps and these are included in Appendix VI.

APPENDIX V

CONVECTIVE COOLING EQUATIONS

The equations for calculating the heat transfer coefficient, h , for a spinning disc with no air cross flow are given below. This flow condition was assumed because the disc is located well inside the wheel and is enclosed by a dust shield on one side and a wheel cover on the other. There is no air flow directed over the surface of the disc and so still conditions were used. The formulae are taken from the General Electric Data Book and are the same as those used by Sheridan, Kutchev and Samie [19].

Assumption: The disc temperature was taken to be constant at 600°C to simplify calculations. The film temperature is the mean of the hot surface and the ambient air temperature and so an approximate film temperature of 300°C was utilised when choosing values for properties of air in the equations. In addition, a uniform temperature over the disc face was assumed for simplicity.

The general equation for these conditions is:

$$h = C_1 \frac{k}{r} \left(\frac{\rho \omega r^2}{\mu} \right)^m \left(\frac{C_p \mu}{0.70k} \right)^a \quad (A1)$$

where C_1 , m and a are constants chosen according to boundary layer conditions and μ is viscosity.

Laminar b.l.: $C_1 - 0.35$ $m - 0.50$ $a - 0.44$

Turbulent b.l.: $C_1 - 0.019$ $m - 0.80$ $a - 0.60$

This is equivalent to $h = C_1 \frac{k}{r} (Re)^n (Pr/0.70)^a$

Reynolds and Prandtl Numbers were calculated:

$$Re = \frac{\rho \omega r^2}{\mu} = 87.9 \times 10^3 \quad (A2)$$

$$Pr = \frac{C_p \cdot \mu}{k} = 0.698 \quad \therefore \frac{Pr}{0.70} = 1 \quad (A3)$$

Assuming turbulent conditions inside the wheel, the equation is reduced to:

$$h = 0.019 \times \frac{k}{r} (Re)^{0.8} \text{ at } v_{max} \quad (A4)$$

As the disc speed reduces throughout the stop, h decreases almost linearly, shown in Fig A5.1. To prevent greatly increased computation time, a mean value was chosen for simulation. This was not the most accurate calculation possible but as this was for demonstration and other significant assumptions had already been made this error was accepted.

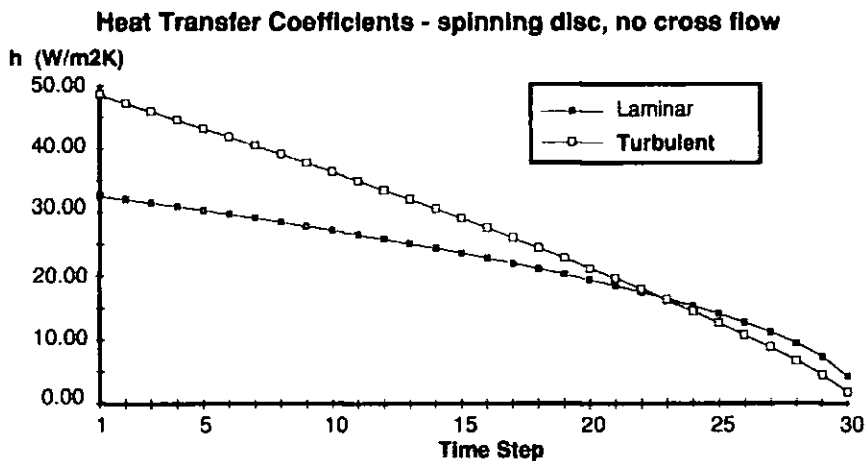


Fig A5.1 Heat transfer coefficients.

APPENDIX VI

FORD EXPERIMENTAL WORK

The following work was completed after the modelling documented in this report. It was carried out on the extensive dynamometer facilities at Ford's Research and Engineering Centre in Essex. It forms the initial stages of work still in progress at the time of writing.

A6.1 Test Facilities

The equipment used consisted of a dynamometer and thermal imaging apparatus. The thermal imaging equipment was fully automated and controlled by computers in the test cell. In addition a runout sensor was mounted on the dynamometer and integrated into the system to allow the disc runout to be correlated against any surface temperature readings. Fig A61 shows the disc and caliper mounted on the dynamometer with the thermal imaging camera on the far left of the photograph. In order for maximum information to be gained from the setup it was necessary to arrange polished steel mirrors on either side of the disc to allow both the edge of the disc and its braking faces to be viewed simultaneously. The 'camera's eye' view can be seen in Fig A62. The thermal image plots later in this Appendix are basically this image processed to straighten out the braking faces that appear curved in the photograph. Hence the plots show the edge of the disc in the centre and rectangular areas either side representing the braking faces. This is illustrated in Fig A64, the positions of maximum runout for each face also being shown. Each individual rectangular plot represents a single disc revolution at a point during the stop.

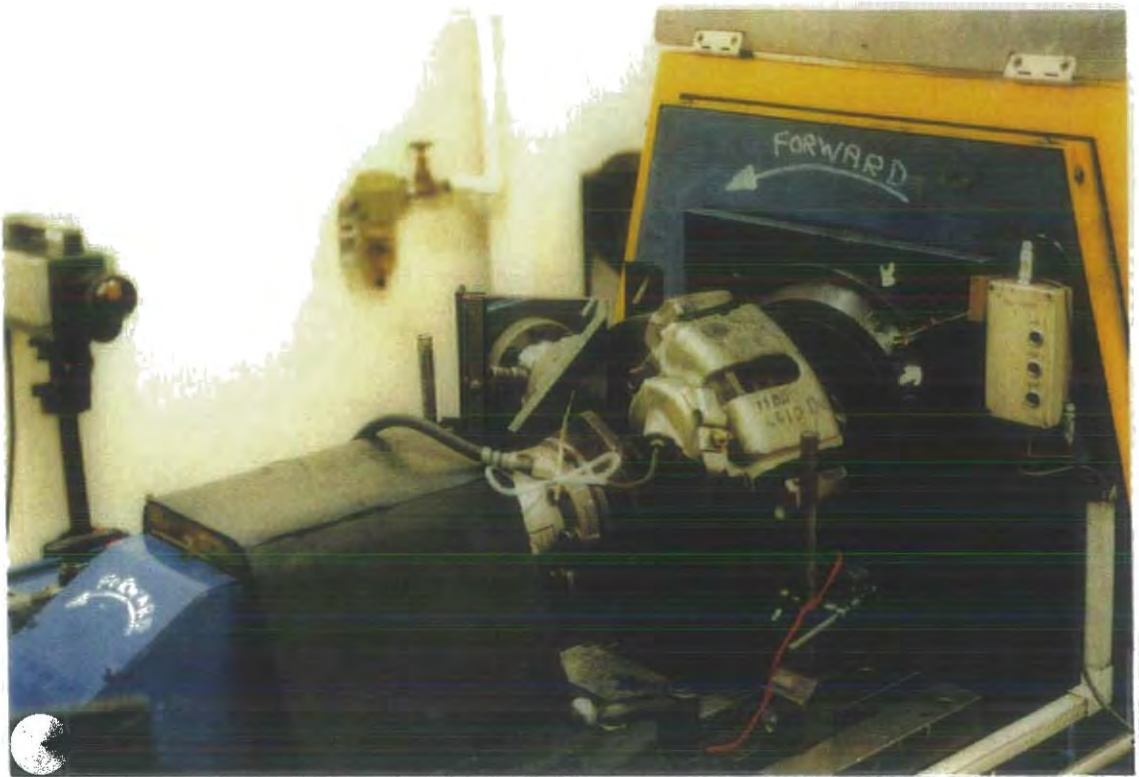


Fig A61 Photograph of the disc and caliper mounted on the test rig.

A6.2 Test Results

Four runs were completed and the results of these runs are shown in Fig A63. The only displacement information available was the location of the maximum runouts for both faces and so no measure of the thermal deformation occurring during the test was available (this will be investigated in subsequent tests). Although temperatures were recorded throughout the test seven individual revolutions are highlighted for each run in the figure. In order to examine the distribution more closely the 774rev in run #2 (second row - last plot) has been shown in Fig A64.

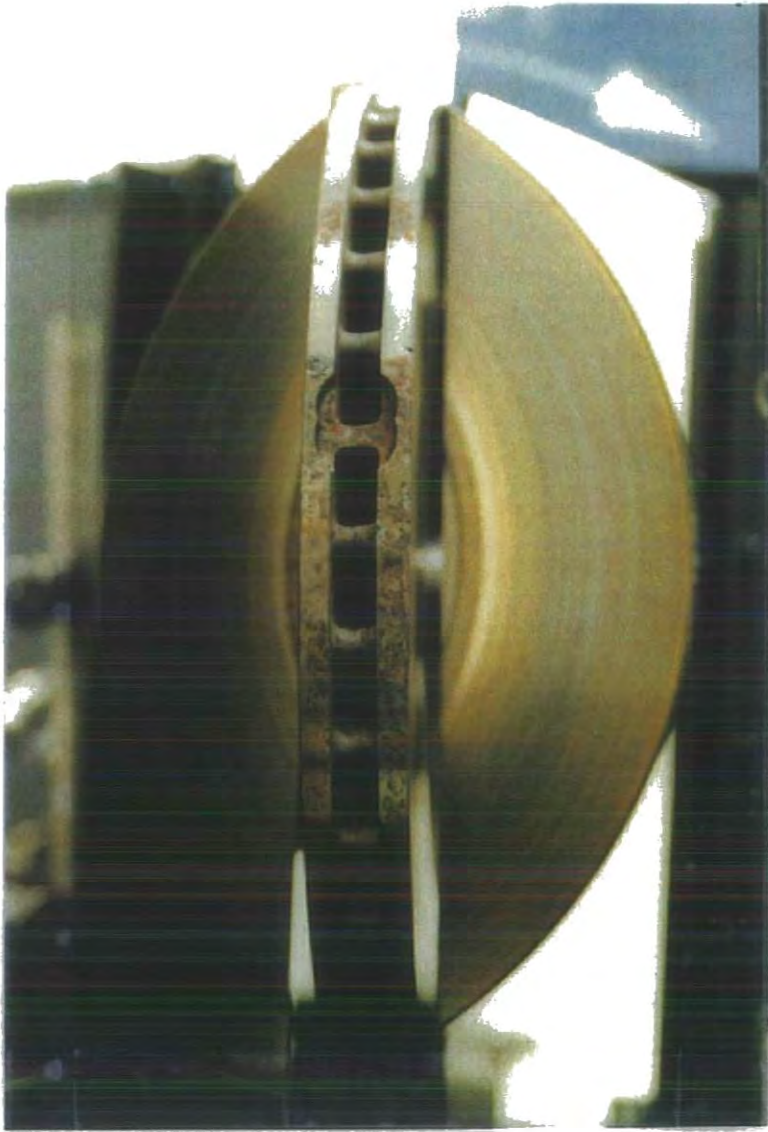


Fig A62 Photograph of the disc view from the camera.

The temperatures in Fig A63 appear to be lower than those predicted by the modelling and it is most likely that such a general difference has been caused by the pad μ value used in the simulations being too high. However it is the pattern of the temperature

distribution that is important. In order to examine this it is necessary to consider Fig A64.

The first and most important characteristic to note is that there appears to be a slight wave in the hot region on each face (pink colour) and this is more pronounced on face #1. Face #1 is at its hottest (the pink bulge in the middle on the left) at its maximum runout point. Similarly the hottest part of face #2 occurs at its runout point at the top of the figure. This confirms the hypothesis that runout causes a 1Hz variation of temperature around the disc and that this will almost certainly result in a variation of thermal deformation around the disc (principally changes in coning causing a wave). This wave will have the same effect as runout and will become increasingly pronounced as the self-excitation takes effect.

In addition the hot region on face #1 is closer to the edge of the disc than the hot region on face #2. This may confirm the effect that pad pressure distribution has on thermal distribution, in this case pad #1 having a large pressure concentration at the outside edge of the disc (see Section 4.1).

The rippling on the boundaries of the thermal contours coincides with the positions of the ribs in the disc. These slight temperature rises are the results of the localised rippling deformation between the ribs, as predicted in the computer simulations and shown on the graphs in Section 5.14.

It is hoped that when further results are available they too provide some confirmation of the observations of the simulations detailed in this report. In addition, it is hoped that areas for further investigation have been suggested.

CDW27 - Disc Surface Temperatures - 4 x 0.87g stops

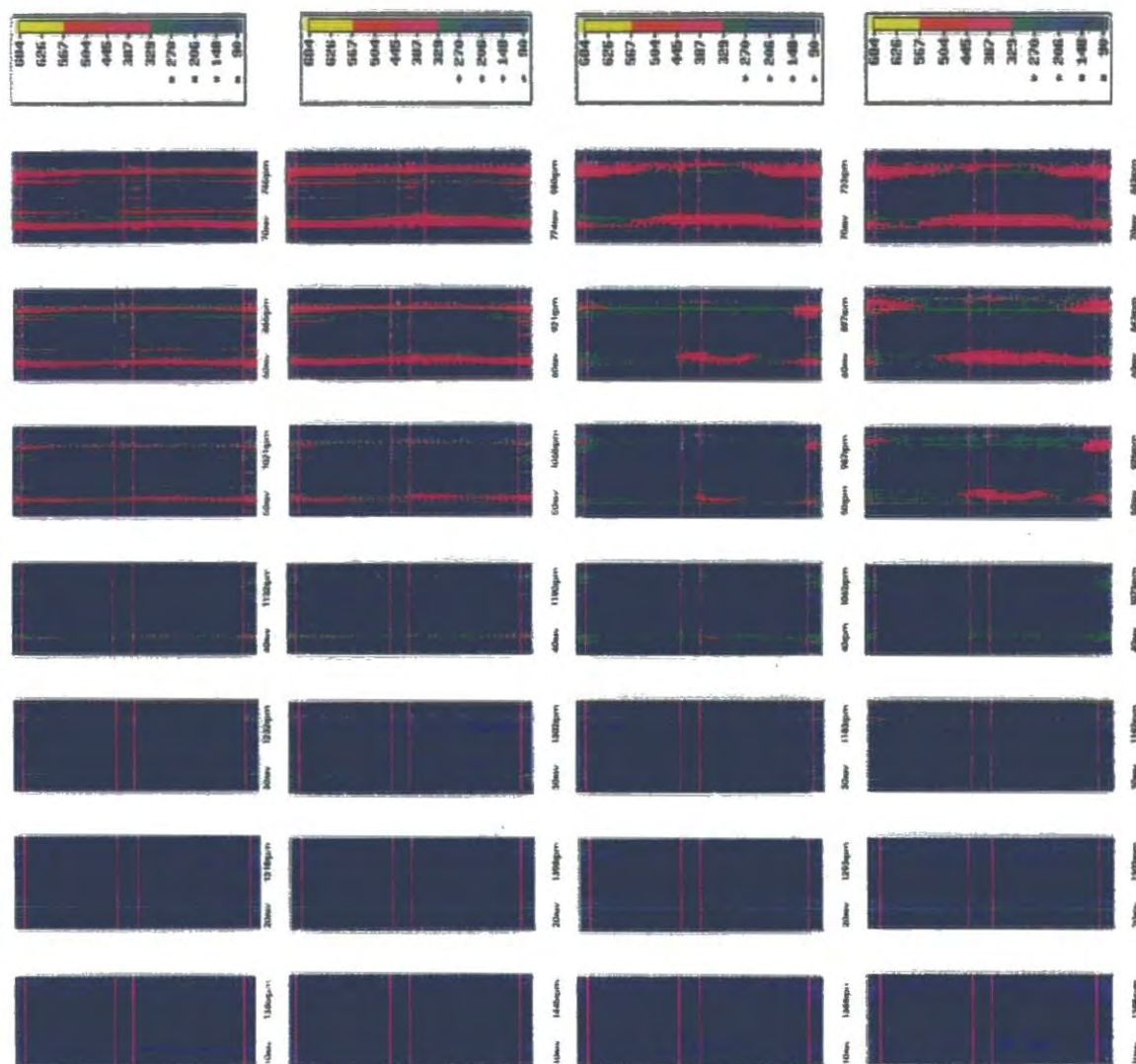


Fig A63 Disc surface temperature maps for four test stops.

CDW27 Disc Surface Temperatures - 0.87g, 774rev.

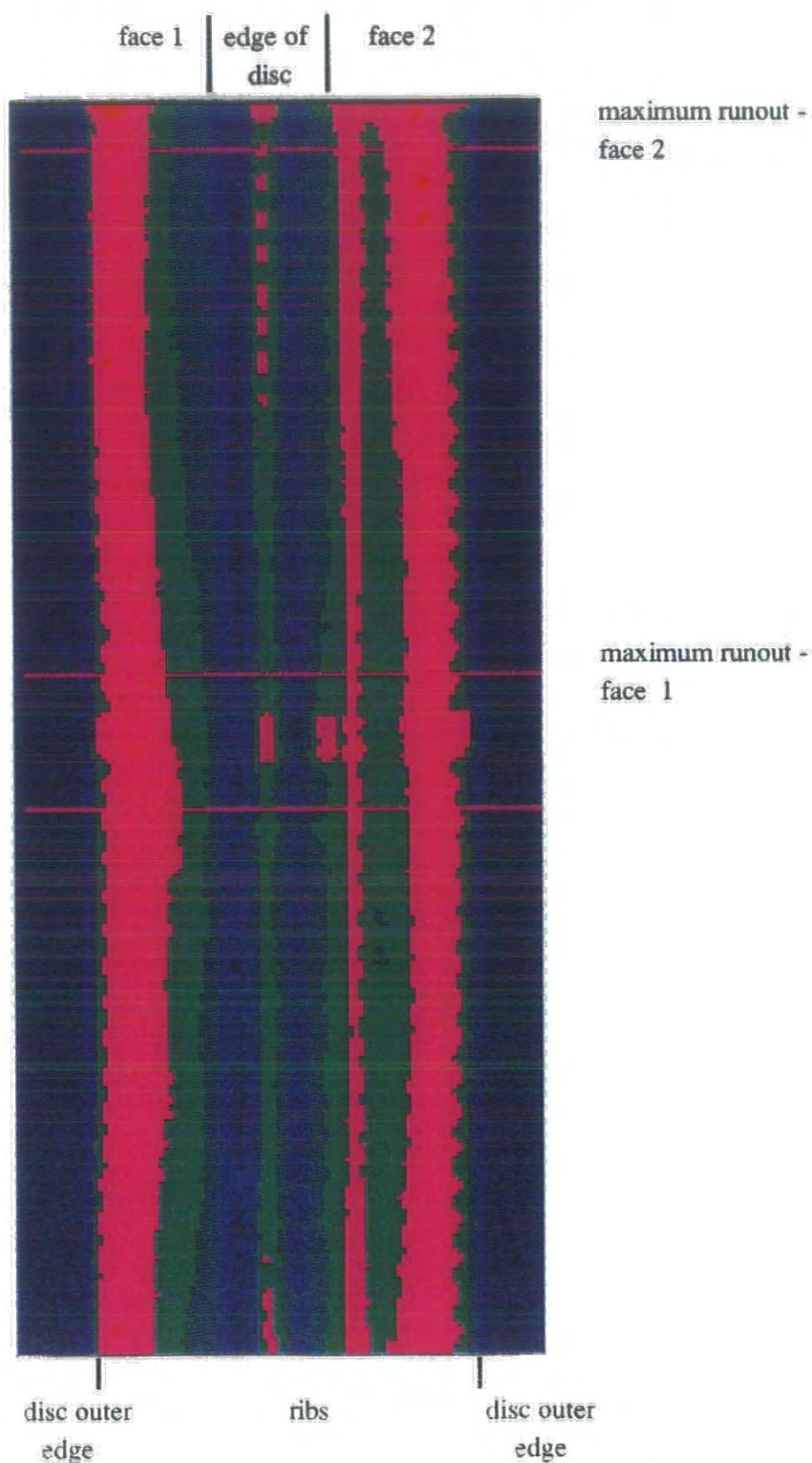


Fig A64 Enlarged temperature map of disc faces.

